POWER

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BUT

By RUFUS T. STROHM

There still are owners throughout the land
Who cling to furnaces stoked by hand;
They quake with dread at the thought of change
And look askance at the new and strange.
Their firemen—this is their boastful claim—
Can put the traveling grate to shame
And beat the underfeed stoker, too.
Well, maybe they can, but they seldom do.

If men in the boiler room took care
To cut off most of the excess air,
And use the damper a little more
Instead of jiggling the ashpit door,
And test each inch of the boiler wall,
And plug the crevices, large and small,
They'd be a mighty efficient crew;
But though they're able, they seldom do.

The stoker works at a measured gait
And runs at a topnotch steaming rate;
Fatigue is a thing that it cannot feel
In its iron frame and its thews of steel.
Yet firemen, trained to the firing game
Insist they're able to do the same,
And make high records the whole year through;
Well, maybe they are, but they seldom do.

Draft Resistance of Underfeed Stokers

By ALFRED COTTON
Research Engineer Heine Safety Boiler Company

THE resistance to the flow of air and other gases, either through orifices or pipes, varies as the square of the velocity when all other conditions remain unchanged.

Various charts have been drawn showing the draft loss through the fuel bed when burning different fuels at different rates. They show that the draft resistance increases much faster than the rate of burning, but

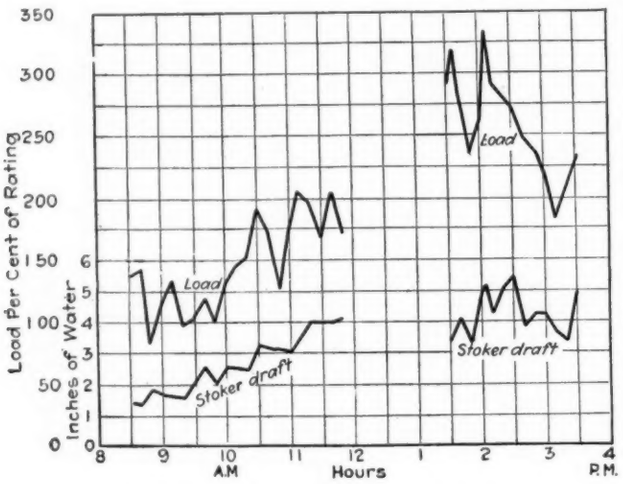

FIG. 1. LOG OF DRAFT AND LOAD

not so fast as the square of the rate. Such curves apply to hand-firing and to chain-grate stokers. As the draft loss through the fuel bed is affected by several things, such as variations in the size of the coal, proportion and fusibility of ash, frequency of cleaning fires and so forth, a great deal of difference in the shapes of the resulting curves is to be expected, though the large averages should show fairly uniform curves following the fundamental law.

Experiments with beds of small shot to simulate beds of burning fuel are described in Bulletin No. 21 of the Bureau of Mines, and the results show close agreement with the law. It is not obvious how temperature would have sufficient effect to cause the deviation shown in the resistance of actual fuel beds; but in view of what follows it must be remembered that the fuel-bed curves at least bend upward, even though they do not do so as much as might have been expected. In some recent boiler trials the departure from the usual curves of fuel-bed resistance is sufficiently remarkable to be worth attention; and as this departure is persistent through a large number of tests the accompanying charts have been prepared to illustrate it. No unusual condition prevailed, and to emphasize this it is advisable to state that the boiler was one of a number of standard 9,500 sq.ft. Heine longitudinal-drum boilers, all equipped with standard 10-retort Sanford-Riley stokers.

Fig. 1 shows the ashpit pressure and load plotted on a time basis. It should be mentioned here that the draft resistance of the fuel bed as plotted is not the actual ashpit pressure, but the difference between pressure in the ashpit and above the fire. The load varied widely and quickly, but the boiler and stoker followed the load so quickly that the steam-pressure variation was slight. The load in the afternoon is nearly double that of the morning, but the stoker draft is not increased three or four times—it is not even doubled.

To show this more clearly the observations have been plotted on a load basis, and they appear as small circles on Fig. 2. The full-line curve is drawn as a mean of the observations. Above 130 per cent load the curve shows that doubling the load requires less than double the draft pressure; and as 300 per cent of rating is passed the draft resistance actually begins to drop.

A large number of tests were made on this boiler for other purposes of research, and to help in presenting this phenomenon the results of the other tests have been plotted for comparison. In comparing them it must be remembered that the circles represent individual observations during a single test, while each of the crosses represents the average stoker draft during an entire test, some of which were of 24 hours' duration. The broken-line curve has been drawn as the arithmetical mean of the averages of all the entire tests, with the exception of the single test of Fig. 1. This curve shows the same falling off of draft resistance around 300 per cent of rating.

Both the individual observations and the averages of entire tests vary considerably from the smooth curves drawn, but not more than is usually the case in such

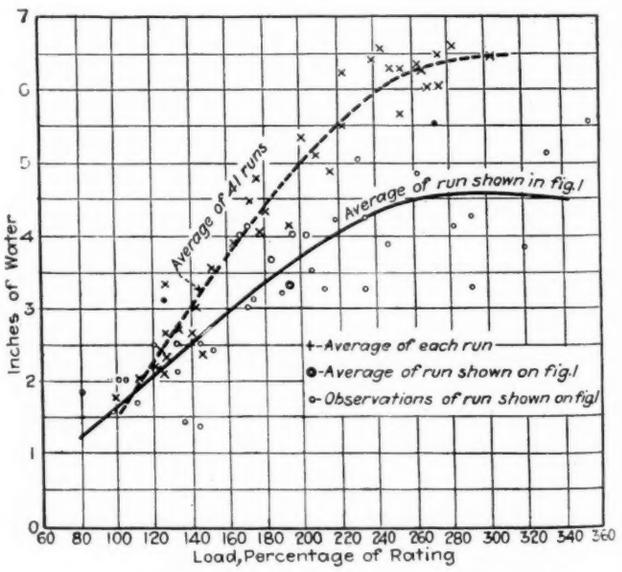
Inches of Water
Average of 41 runs
Average of run shown in fig.1
+-Average of each run
•-Average of run shown on fig.1
o-Observations of run shown on fig.1
Load, Percentage of Rating

FIG. 2. RELATION BETWEEN DRAFT AND LOAD

observations. These deviations are caused by the condition of the fuel bed, whether the load is rising, falling or stationary at the time the observation is made, amount of ash and clinkers and so forth. To facilitate accommodating the curves to the observations they were drawn to a much larger scale.

The writer suggests that the departure of these curves from the usual fuel-bed resistance curves is due to the

method of firing being quite different from either hand or chain-grate firing, where comparatively little disturbance of the fuel bed occurs. In these underfeed stokers, as the rate of coal feed increases the disturbance of the fuel bed increases also. As the rate of driving approaches 300 per cent of the rating of the boiler this

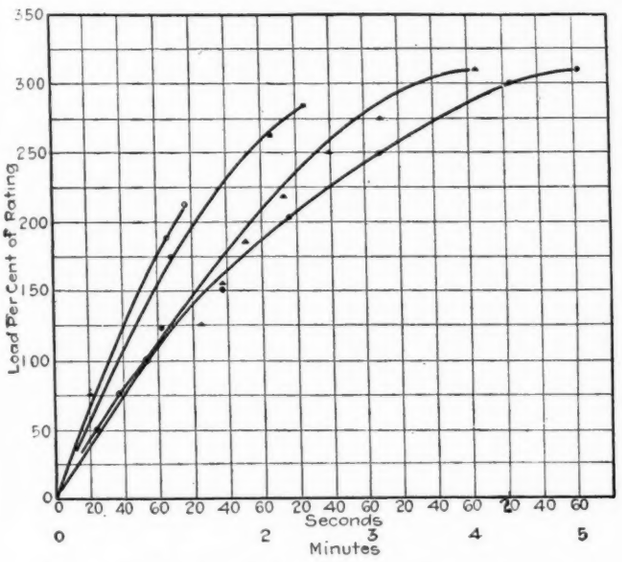

FIG. 3. RATE OF LOAD INCREASE STARTING FROM BANKED FIRES

disturbance is probably so pronounced that caking of the coal and formation of clinker is impeded to such an extent that the resistance of the fuel bed is much decreased—so much so that the draft loss actually begins to decrease with increase of load.

It will be noticed that the broken-line curve based on all of the entire tests is much higher than the full-line curve based on the single test. The only difference between these tests is that the test of Fig. 1 was made with a widely varying load, while the rest of the tests were run with a fairly constant load. Following the foregoing line of reasoning, this would seem to suggest that frequently varying the amount of disturbance is more effective in reducing draft resistance than a constant disturbance. After a quieter period which had allowed some caking of coal and fusing of ash the effect of an increased disturbance would be greater than the effect of a continued large disturbance which prevented much caking and fusing. That is to say, when some clinker has formed the effect of an increased disturbance would be more widespread.

The coal used was a mixture of several Pennsylvania bituminous coals that had been stored for a considerable time. It averaged around 13,500 B.t.u. in heating value and clinkered very easily.

The average load and air pressure of the test shown in Fig. 1 are represented in Fig. 2 by the larger circle.

Neither of the curves originates with no pressure at no load. The chimney effect of the fire might have some influence. Also, it is probable that at less than 100 per cent of rating, when the disturbance of the fuel bed would be small, the curve may be similar to the usual fuel-bed resistance curves, and this would carry those drawn in Fig. 2 toward an orthodox origin.

It may be contended that in drawing draft-loss curves the mean of the square roots of the pressures observed should be used rather than the arithmetical mean; but as the curves follow actual velocities more closely than their squares, the arithmetical means have been used throughout.

It may be reasonably suggested that if the offered solution is correct the amount of excess air at the higher driving rates was probably very large. This was not so, however, for the percentage of CO_2 in the flue gases gradually and consistently rose with increase of load, checked with the heat balances which show a somewhat higher efficiency characteristic for the higher loads than is usually the case.

In reading Fig. 1 there may be some uncertainty concerning the lag in following changes of load. Tests were run to ascertain how quickly the boiler and stoker would respond to sudden large load changes, and Fig. 3 shows a few curves of the time taken to pick up heavy loads from a dead-banked fire. The loads were measured by a General Electric flow meter and therefore show steam actually generated and sent into the main steam line. The longest time taken to shoulder a load of 300 per cent of rating when starting from a dead bank was 4 minutes 23 seconds, so it is evident that no lag of any magnitude need be considered in examining Fig. 1.

Vortical Process of Combustion

In the generation of steam for power the action of boilers depends greatly on the method with which fuel is applied. The velocity of the gases through the boilers is approximately 2,400 ft. per min. and the distance from stoker to the nearest boiler tube has been in the past about 5 ft., is now 10 ft. and over, and in the future probably will be 20 ft. It can readily be seen

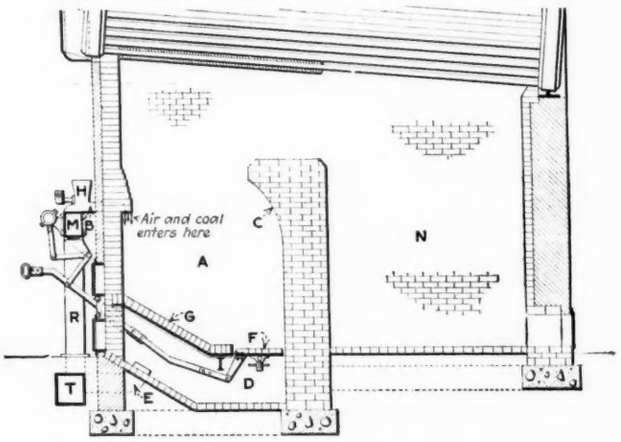

FIG. 1. DIAGRAM OF FURNACE

that at a velocity of 40 ft. per second, with a distance of 10 ft. from stoker to the tube, combustion should be completed in one-quarter of a second. These figures bring out the necessity for proper combustion in the shortest period of time possible. From the foregoing it may be concluded that if any other method be developed by which the period of time that the gases take when traveling from the fuel bed to the tubes is greatly increased, or if the time element is eliminated, then more complete and efficient combustion will be obtained without raising the boilers to a greater height. One way would be to burn coal in suspension.

Burning coal and other solid fuels in suspension has been accomplished for several years in an experimental

plant located on the property of the United States Bureau of Mines at Pittsburgh. This plant is owned by the Stephens Engineering Co. Phillips Badenhausen has taken over the future engineering in connection with the development of this system of firing, as well as manufacturing and selling. The furnace is now known as the Phillips Badenhausen-Stephens furnace for use in boilers and it is claimed that it will burn solid fuels of a half inch or less in suspension, whether the coal be bituminous or anthracite, lignite, anthracite culm, coke braize, rice chaff or sawdust, etc.

The furnace consists of a firebrick chamber *A*, Fig. 1, with a bridge wall *C* so constructed as to deflect the pieces of fuel carried upward along in surface toward the opposite side of the firing chamber, so that in operation the furnace has the appearance of that shown in Fig. 2. Under the furnace bottom is an air chamber *D*. Next to the bridge wall is a movable element *F*, which forms part of the furnace bottom and is pivoted so that it can be turned from the horizontal to a vertical position, thus constituting a dumping platform. The air duct *T* is located below the boiler-room floor and communicates with the air chamber *D* through the connection *E* and through the upright column *R* to the manifold base *M* of the blast tubes *B*. Along the outside of the furnace there are a number of rotary coal feeding devices placed above the outer end of the blast tubes *B*.

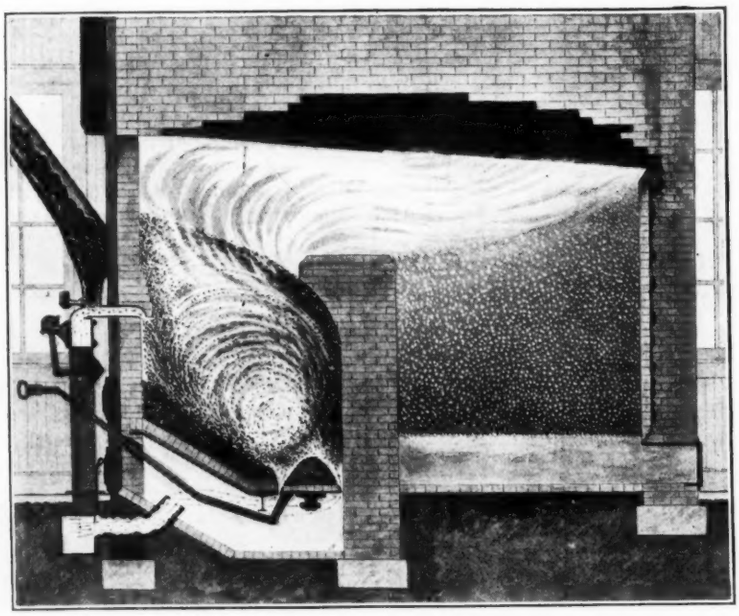

FIG. 2. SECTION THROUGH FURNACE WHEN IN OPERATION

Fuel is introduced at the front of the chamber, mixed with air and is cast downward, building up and thereafter maintaining an incoherent, incandescent bed of fuel particles on the furnace bottom. In the downward movement of the fuel the dust and fine particles are burned before reaching the bed. The action of the coal and air is such, combined with the structure of the bed, which is built up below the angle of repose, that the bed flows by gravity toward the opposite side of the chamber, all the particles of the bed being in a disassociated condition and each particle being coke of a consistency of dry beans. Upon the fuel bed reaching the opposite side of the firing chamber, the overflow is met by an upward stream of air, from chamber D, which carries the overflow into suspension until it is deflected back toward the front of the chamber and from there back to the bed, thus producing an active vortex in which the coke content of the coal is carried.

It is claimed that while in the vortex the fuel is burned on the outside, and ash is formed upon the surface. This ash is thrown off and, being too light to maintain itself in the vortex is carried over the bridge wall and deposited in a dead air space *N* in the form of a light brown powder. The ash given off by this process saves the heat and radiates it to the boiler tubes, while any flakes of carbon in the ash are burned so that only a powder remains. All the smoke is consumed, as is also the combustible. Such percentage of the fuel as that made up from rock and slate drop to the lowest point of the firing chamber and are removed by dumping the movable element *F*.

At the present time the Phillips-Badenhausen-Stephens furnaces are being put into commercial operation at a plant containing three Badenhausen boilers having 5,000 sq.ft. of heating surface. It is expected that within six months all practical tests will be completed and that the results obtained from this furnace in actual commercial operation using anthracite and bituminous coal without change of furnace design, will be available for publication.

A representative of *Power* visited the plant located on the property of the United States Bureau of Mines at Pittsburgh, and had an opportunity to see the Badenhausen-Stephens furnace in operation burning bituminous screenings. Although tests were not made so that an intelligent opinion could be passed upon the process, nevertheless so far as the furnace was concerned the indications were that good combustion was being obtained and that the firing and furnace temperature could be readily controlled. During operation the boiler was supplying no load, the steam generated being blown off through a muffler. Under all conditions of firing smokeless combustion was obtained.

Engineers' Salaries

Some interesting information is brought out by a compilation of the average income of the class of 1910, Cornell University, recently made by its class secretary. Of the 372 members of the class 83 (more than any other vocation) were engineers, with an average yearly salary of $3,724. The only average salary lower than this was that of teachers, 33 in all, with an average of $3,137. Of the 17 vocations listed, the highest paid was that of bankers and brokers, 10 of whom received an average salary of $11,040 per year. The average for all members was $5,385, including veterinarians, farmers, accountants, salesmen, architects and physicians. The average salary of advertising men was the nearest to the average for the entire class.

High- and low-water alarms have their advantages but also their disadvantages. In the opinion of many engineers they should not be used, as attendants invariably come to rely solely on their automatic action, with the result that frequently the water level runs dangerously low through failure of the alarm.—*Warren Hilleary.*

Important Features of Control Apparatus*

Contactors and Contacts—Three General Types of Contacts—Temperature of Controller Contacts—Blowouts and Arc Boxes—Construction of Controller Coils

By H. D. James
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In control engineering the development has progressed rapidly during the last few years. This branch of engineering is becoming standardized for ordinary applications. Many of the difficulties have been eliminated by careful study, with the result that modern control apparatus has become more reliable and less complicated. Often, the service requirements do not warrant automatic controllers, and where these controllers are more expensive than those manually operated, it is not good business to use them, although they may have greater durability and added convenience. Alternating-current circuits of 2,200 volts are now fairly common, and the question frequently arises as to whether air-break or oil-break apparatus should be used. These and other questions will be brought out in this and a subsequent article, and as far as possible the author will indicate the advantages and limitations of various constructions.

Contacts and Contactors

The most important feature of any controller is the contact. The function of the contact is to carry current and to withstand the abrasive action when rupturing current. Sometimes the contact is made up of two parts—the main current-carrying part, such as a laminated brush, and the arcing portion, which is subject to abrasion. Controllers are essentially different from switchboard apparatus in that the durability of the contact and other parts is of greater importance in many cases than the ability to carry current. Most motors are started and stopped at frequent intervals and sometimes under heavy overloads. Even motors used for driving fans and blowers for ventilating purposes may operate only a part of the day. On the other hand, generators, transformers and feeders connected to switchboards may remain on the circuit for weeks at a time, so that the accumulated heating in the current-carrying parts of the switchboards, particularly the contacts, is of the greatest importance.

A contactor is for the purpose of opening and closing the electric circuit to a motor repeatedly under normal operating conditions. A circuit breaker is for the purpose of continuously carrying the current and of interrupting this current under abnormal conditions, usually very infrequently. The contactor is expected to rupture the circuit many thousands of times without repairs. Oil circuit-breaker engineers have stated that a circuit breaker may open the circuit two or three times under abnormal conditions and still remain operative. This distinction should be borne in mind in order to obtain the proper viewpoint in discussing controller contacts.

Contacts are usually made of copper. Some of the smaller contacts are either graphite to graphite or graphite to copper. Very small contacts frequently have one element made of graphite, as less pressure is required on such a contact for carrying small currents. The current-carrying capacity of a contact depends upon the heat that it will absorb and radiate. It is, therefore, dependent on the mass and radiating surface and also on the surrounding current-carrying parts that generate heat. Contacts do not ordinarily have area. Two flat surfaces brought together make contact only at a series of points. Many contacts have cylindrical surfaces and should be considered as having line contact rather than area contact. The wider the contact within certain limits, the more current it will carry. In pressing two copper surfaces together, a certain minimum pressure must be used per lineal inch in order to obtain reasonable contact between the two surfaces or, in other words, to mash down the points and increase the number making contact. As this pressure is increased, the current-carrying capacity also increases owing to the more intimate association of the two surfaces. Therefore, within certain limits increased pressure results in increased current capacities.

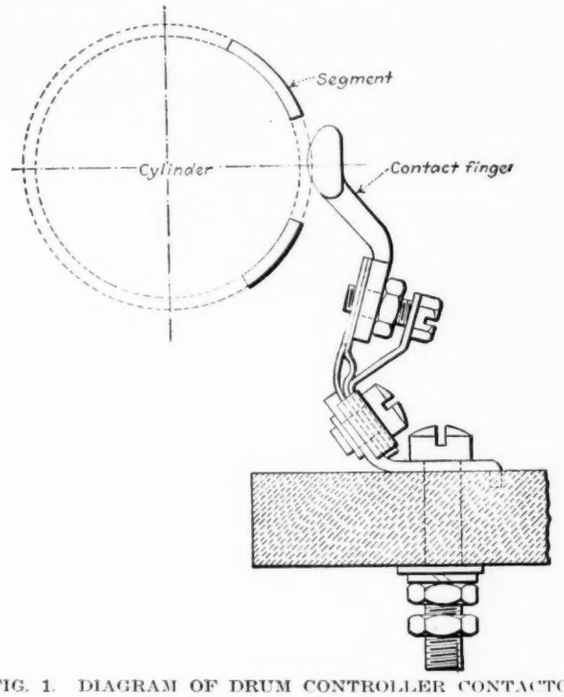

FIG. 1. DIAGRAM OF DRUM CONTROLLER CONTACTOR

There are three general types of contacts—butt contacts, sliding contacts and rolling contacts. A good example of butt contact is the laminated brush of a

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large circuit breaker, where the leaves of the brush engage the copper plate normal to the surfaces. The theory of this brush is that each lamination makes a line contact with the copper plate, and therefore the total contact is the sum of these lines multiplied by their width.

The sliding contact, sometimes called a wiping contact, is well illustrated in the drum controller, Fig. 1. In this case the contact finger is stationary and engages the copper surface of a revolving cylinder. Some types of magnetic contactors, particularly the smaller ones, are designed so that, in closing, the movable contact slides on the surface of the stationary contact. When the contacts are both made of copper, experience has shown that if the pressure per inch width of contact exceeds ten pounds, the cutting action becomes pronounced and the surface should be lubricated in order to keep down mechanical wear.

Action of Rolling Contact

The rolling contact, Fig. 2, is the most successful type now in use. As usually made, it consists of two L-shaped pieces, the contact surface of the L's being bent in a curve. As the contacts approach, they engage at the tip, or outer edge of the curve, as at *B*, and roll down to the heel, or portion next to the angle, as at *C*. In this way the burning at both closing and opening takes place at the tip and away from the current-carrying part.

When two copper contacts are moved together, either by a magnet or by hand, there is a tendency to bounce at the instant of engagement, owing to the elasticity of the contacts. This bounce depends upon the speed and is greater in magnet-closed contacts than in hand-closed contacts. When a heavy-duty contactor closes, a slight arc can usually be observed, which tends to soften the contact surface. If contact is re-established on the softened parts of the surface, the contacts may

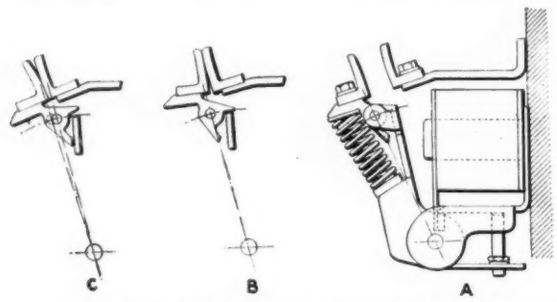


FIG. 2. SHOWS THREE POSITIONS OF A MAGNETIC CONTACTOR, ROLLING-CONTACT TYPE

weld; this welding always occurs at the time of closing and not at the time of opening. When the bounce takes place with the rolling contact, it is rotating around a center, which in turn is moving with considerable speed toward the closed position. When contact is re-established after the bounce, the engagement takes place farther down, on a different part of the contact, so that the softened portions of the contacts do not come together. If the contacts are opened and closed very rapidly, the accumulated heat has a tendency to soften a considerable part of the contact surfaces near the tip or point of engagement, so that a slight sticking action may take place; but the action of the rolling contact exerts a very great leverage for separating the two surfaces as they roll from the tip to the heel, so that a relatively small closing force can separate an appreciable weld. The result has been that the rolling contacts seldom weld if properly designed. With butt or sliding contacts the only force available for separation is the direct pull. It is undesirable to have much sliding motion combined with the rolling action. All contacts of the rolling type have a very slight sliding motion, but any considerable slide on the roughened surface of the contacts may lock them together in extreme cases and prevent the magnets closing the air gap. Under such conditions, if an alternating-current magnet is used, a much heavier current will pass through the coil and may burn it out.

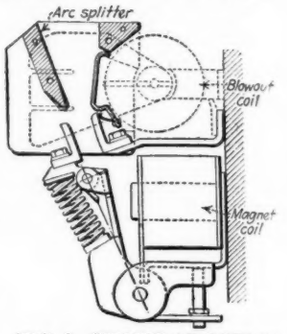


FIG. 3. SHOWS ARC SPLITTER ON CONTACTOR

The temperature of controller contacts is a much discussed subject. Differences of opinion arise from not stating definitely the kind of contact under discussion. If a laminated contact or a knife contact is considered, which depends upon the resilience of the copper, it is evident that the temperature must be kept low, so as not to impair the temper of the copper. This usually applies to the contacts used on switchboards. Where the current-carrying function of the contact is the controlling element in the design, there is no difficulty in using brush or laminated contacts, but where arc rupturing is an important consideration, the heat produced by the arc may be very much in excess of the heat produced by the actual carrying of the current, so that the temperature of the contacts may be raised above that which would draw the temper in the copper leaf.

Contacts of this kind are usually made of solid metal, and the resilient action is obtained by a spring located remote from the arcing surfaces. A solid-copper contact of the rolling type may be worked for years at a relatively high temperature without in any way reducing its ability to carry current. If an investigation is made of the temperature of the contacts on a crane or hoist, it will be found that the heating due to the effective current is much less than that due to the arc. It is, therefore, an error to judge such a contact only by its current-carrying ability. If a brush contact is used, it should have an auxiliary arcing contact in order to protect the brush from arcing.

If a careful test of the voltage drop through the different parts of a contactor is made, a millivoltmeter will show that the energy lost in the blowout and shunts represents the largest part of the loss and that the drop across the contact is usually less than 10 per cent of the total drop. Tests show that the heating from the contact itself is not great, but that a good deal of the temperature is caused by the heating of the other parts. A contact without blowout will run much cooler than one with blowout. This becomes more and more a factor as the size of the contact increases.

Blowout and Arc Box

A magnetic blowout is the usual method of rupturing an arc in a controller. In this a magnetic field is projected across the path of the arc, which is a flexible conductor, and behaves in the same way that any other

conductor would in a magnetic field; that is, it moves through that field in the same direction as a conductor on a motor armature would move in a motor field. The length of this arc is a function of the voltage; this voltage is the sum of the initial line voltage of the circuit opened and the induced voltage caused by the stored energy in the circuit, commonly called the inductive kick. For medium-sized controllers connected to series motors, this inductive kick may be assumed to add approximately 50 per cent to the line voltage when the circuit is ruptured by a single contactor. Usually, the feeder supplying power to the motor is relatively short, and therefore the inductive effect is nearly all caused by the motor reaction. The diameter of the arc is in proportion to the current, and the arc itself is an elastic conductor having a core or central portion which carries the current, and an envelope of vapor surrounding this core. The diameter of the core is automatically increased to correspond to the current. The temperature decreases from that of the core to the air temperature at the outside. The flash that is observed extending beyond the arc box when a circuit is ruptured may be only the vapor envelope and not the actual current-carrying core. Arc boxes should be made wide enough to provide considerable space between the core and the box sides. If the box is too narrow, the high temperature of the core may come in contact with the sides and cause rapid abrasion.

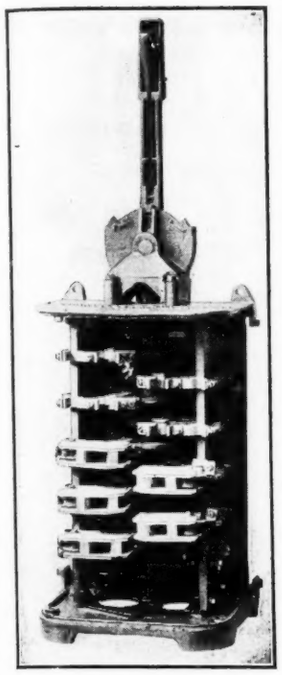


FIG. 4. CAM-TYPE MANUAL CONTROLLER WITH COVER REMOVED; SUITABLE FOR USE ON CRANE AND HOIST DUTY

Rupturing an Electric Arc

In order to rupture the arc, it is necessary to stretch this elastic conductor and at the same time cool it. The quicker it is stretched and cooled to the rupturing point, the less the wear on the contacts and arc box. A common method of stretching this arc is to blow it out in the form of a loop from the ends of the contacts. The conductor will travel rapidly as long as it is within the magnetic field, but if it is not ruptured before it reaches the edge of the field, it may hang on for an appreciable length of time before it is cooled down to the rupturing point. If a barrier is introduced in the path of this loop, two loops will be obtained, which have a total length within a given magnetic field that is much greater than the length of a single loop. Also, the cooling effect of this barrier helps to extinguish the arc. These barriers are often called arc splitters, Fig. 3. When properly located, they materially reduce the time required for rupturing the arc and also reduce the flash external to the arc shields (see Fig. 5). Stated in another way, with constant voltage and a given size arc box and magnetic field, the arc can be extinguished much quicker with an arc splitter; or the use of an arc splitter enables the contactor to open a higher voltage arc while it is still within the magnetic field.

Fig. 4 shows a manual controller with the cover removed, suitable for crane and hoist duty. It consists of a central shaft on which are mounted a series of cams which actuate contactors. These contactors are the same as magnetic contactors except that they are opened and closed by means of a cam instead of a magnet. The contacts are of the rolling type, and the arc boxes are provided with arc splitters similar to those in Fig. 3. The arc boxes and contacts are interchangeable with coresponding sizes of magnetic contactors.

Where the line voltage does not exceed the 550-volt class, a single arc splitter is used, but where the magnetic contactor ruptures higher voltages, such as 2,200 or 6,600 volts, a series of arc splitters are used, breaking the arc up into a number of loops and interposing cooling barriers in each of these loops. In using arc splitters, the area between the arc splitter and the sides of the box must be kept large enough to accommodate the current stream representing the maximum requirements of the contactor. This requires a larger arc box to accommodate these splitters and limits the number of splitters that can be economically used.

Two Contactors May Be Used in Series

The arc stream consists of small charged particles called ions, which travel rapidly from one contact to the other; these ions are blown out by the magnetic field. But there are additional ions which are not in the arc stream and remain between the contacts. The voltage between the contacts establishes a static field which tends to set these ions in motion, and they may re-establish the arc. An improperly designed contactor will permit the arc to be re-established two or three times before it is finally ruptured. As the switch opens the farther the contacts are separated the weaker becomes the static field, which reduces the liability of these inactive ions re-establishing the arc. Where the opening of the contactor is not sufficient to prevent re-establishment, two contactors may be used in series. This greatly reduces the static field and also distributes the energy between the two arc boxes.

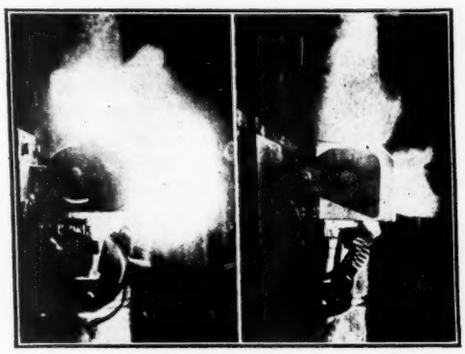


FIG. 5. TWO MAGNETIC CONTACTORS; ONE ON RIGHT EQUIPPED WITH ARC SPLITTER; ONE ON LEFT WITHOUT ARC SPLITTER

The arc box should be made of refractory material without a fusible binder. The shape of the magnetic field should be laid out carefully, as the flashing of the arc to other parts of the contactor may be caused by an improperly shaped magnetic field. Where arc splitters are used, this field should be shaped to take full advantage of the splitters.

Controller coils are usually shunt or series wound. Sometimes compound coils may be used, but this occurs to a very limited extent. For a time a prejudice existed against a shunt-wound coil with fine wire, because of the breakage of this wire. The terminals also presented a weak point, the lead wires often pulling loose from the winding. The construction of shunt coils has been improved very much; many of these coils are now wound on automatic machines which stop when the proper number of turns have been placed on the coil. The machines also distribute the winding evenly in layers so that the same ohmic resistance is obtained in all coils wound from one setting of the machine. Enamel wire is used extensively, and the automatic machines have been arranged to weave cotton between the layers of the wire and form a washer of cotton at each end of the coil. The coils are then impregnated and given a heat treatment, followed by a varnish covering. This process makes of the coil a solid mass which is not injured by vibration and radiates heat very readily.

Temperature of Controller Coils

Temperatures at which coils can be operated are no longer limited by cotton insulation. Heat treatment increases the melting point of the gum so that coils of this kind have been operated successfully at 165 deg. C. measured by resistance for over 2,000 hours. The impregnated washer at each end of the coil seals the joint between the insulating sleeve inside the coil and the end washer, which prevents insulation breakdowns at this point. Improvements have been made in attaching the lead wires, one method being to attach the end of the coil winding to a terminal clip which is impregnated fast in the insulation surrounding the coil and extends from the coil only far enough to attach a wire. This does away with the flexible leads and makes the coils easier to handle.

Coils should be impregnated with a gum that is slightly elastic so that it will yield at the higher temperatures and adapt itself to the expansion and contraction of the windings. If the impregnated compound becomes hard and brittle, it will form minute fissures in the coil, which absorb moisture and cause insulation breakdowns. The varnish treatment that the coil receives after impregnation is impervious to moisture when it is intact, and renders the coil highly moisture resisting. In time these outer coatings wear down, and unless renewed, the coil becomes more and more affected by moisture. It is, therefore, undesirable to subject such coils to the direct action of water, although the occasional wetting of the coil with spray will not result in injury.

Contactor Coils Versus Motor Coils

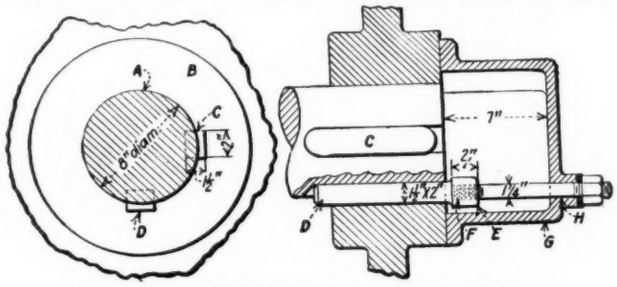
Heat distribution in the coil of a magnetic contactor is different from that in a motor. The shunt-field coils of a motor are ventilated by the revolving armature and cooled by the core. In fact, the only heat received from this core comes from the coil. The coil in the magnetic contactor is not artificially ventilated, and the iron core inside the coil receives some of its heat from the current-carrying parts of the contactor.

In order to study the distribution of heat, special coils were built with thermocouples in the winding at different points. Heat tests were made and the temperature taken by these thermocouples, also by a thermometer on the outside of the coil, and by the resistance of the coil. When the resistance method was used, the hot-spot correction was approximately one degree for every one-tenth inch depth of winding. Sufficient experience has not been obtained to state that this relation exists in all designs, but it represents a close approximation and is sufficiently accurate for practical purposes, the usual factor of safety in windings of this kind being sufficient to take care of small differences. The thermometer readings obtained on various tests showed no consistency and were very misleading. When the temperature is measured by a thermometer placed on the outside of the coil, the reading of the thermometer will be somewhat less than that obtained by the resistance method. The standard allowance made by the A. I. E. E. is 5 deg. C.

Starting and Extracting Stubborn Keys

A correspondent writes in *Practical Engineer*, London, as follows:

My plan in fitting keys to the ends of large shafts and wheels is to provide both a feather and a key, the feather to "drive" only, and the key to hold the wheel "mainly," and to drive as an aid to the feather. The key has an enlarged head, which is drilled to take a 1¼-in. stud before it is driven in for the last time. To drive



HOW A STUBBORN KEY CAN BE REMOVED

it in finally, a piece of brass is used so as not to injure the thread. To extract it is an easy matter, and the tightest key may be started in a few minutes.

In the sketches *A* and *B* are the 8-in. shaft and wheel-hub, *C* is the feather and *D* the driven key. *E* is the enlarged head of the key *D*, while *F* is a 1¼-in. diameter tapped hole. *G* is a strong bridge piece, while *H* is a 1¼-in. stud. To draw the key, place the bridge across the face of the hub, insert the stud and then pull up on the nut.

I have seen keys that would not start by this method, but I have succeeded in this way: Having pulled up the nut as tightly as was reasonably possible with a 2-ft. wrench, I have heated the stud all along the exposed part to a good heat with a gas torch. The heating of the stud expanded it in length, and then the nut was pulled up tightly again and the stud allowed to cool off. While cooling of the stud takes place, a shrinkage strain comes on the stud, on the bridge and on the key, and I have known keys to start suddenly from their seats in this way on more than one occasion.

A 32½-mile broad-gage railroad line is to be built in California by the Pacific Gas and Electric Co. simply to carry the materials needed in the construction of the Pit River power plants. The road will start at the end of the McCloud River Railroad Co.'s line, at Bartel, and will run through Cayton Valley to the site of Pit River House No. 1. It is estimated that this method of haulage will be cheaper than the use of trucks.

Steam-Turbine Shaft Glands

By JOHN R. BAKER

STEAM turbines that exhaust into a condenser in which a high vacuum is maintained must have means installed to prevent leakage of air into the turbine where the shaft passes through the casing, or shell. The economy of the turbine, or the pounds of steam used in an hour for each horsepower developed, is very much improved by increasing the vacuum, for instance, from 26 to 29 in. The pressure of air in the condenser throws additional work on the air-removal apparatus attached to it, and renders the high vacuum desired difficult of attainment. In one instance a one-inch pipe plug located in the wall of the inlet of a large surface condenser became loose and fell out, and before the leak was found and stopped, a matter of only a few minutes, the vacuum dropped from 28.5 to 17 in. If the vacuum breaker on a condenser can be opened a minute fraction of its travel, the vacuum gage may be seen to drop four or five inches with the inrush of air through the valve scarcely audible. To maintain the turbine economy at the highest point it is therefore necessary to keep the air leakage into the turbine and condenser at a minimum, or no more than the non-condensable gases brought in by the boiler steam.

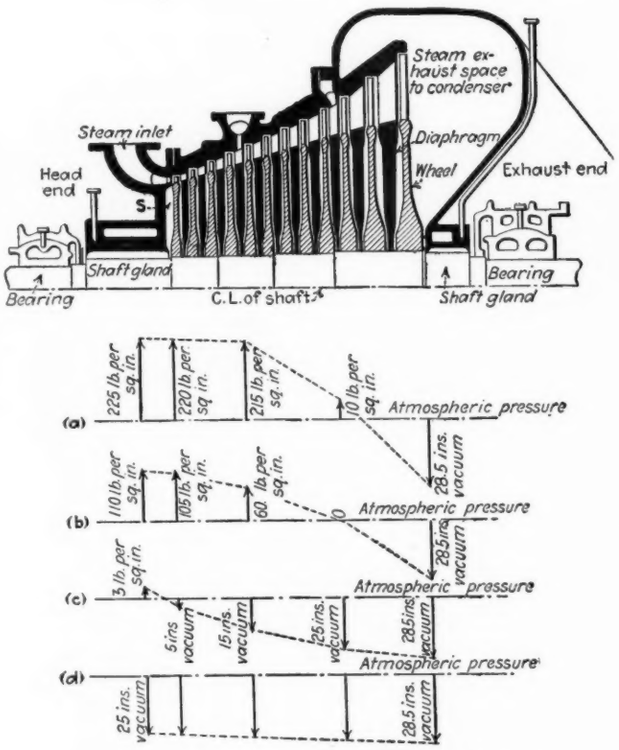


FIG. 1. CROSS-SECTION OF TURBINE AND PRESSURE DIAGRAMS

When running at speed and carrying some load, air leakage into the shell may occur only at the exhaust end of the turbine. The pressure here is negative, or less than atmospheric, and equal almost to the vacuum in the inlet of the condenser. The vacuum of 27 to 29 in. inside the turbine casing produces a suction along the shaft, pulling in air from the outside. This is not the condition at the other end of the turbine, sometimes called the head end, where steam is being admitted. The steam pressure in the head, in turbines governed by throttling, will vary with the load carried by the turbine, but will always be greater than the atmospheric pressure when any load of amount is on the turbine. The direction of leakage will be outward along the shaft, and steam will escape, unless restrained, into the atmosphere. These conditions are represented in Fig. 1, showing a half-section along the shaft of an impulse-type turbine. Below are four sets of lines to represent the pressure throughout the turbine under (*a*) full load, (*b*) one-quarter load, (*c*) running unloaded and not up to speed, and (*d*) with a standing shaft and the throttle valve closed. The direction of steam flow is seen to be along the shaft, while the pressure decreases from the inlet pressure at the head end to the vacuum at the exhaust end. Under load conditions, as shown by (*a*) and (*b*) leakage is outward at the head end and inward at the exhaust end.

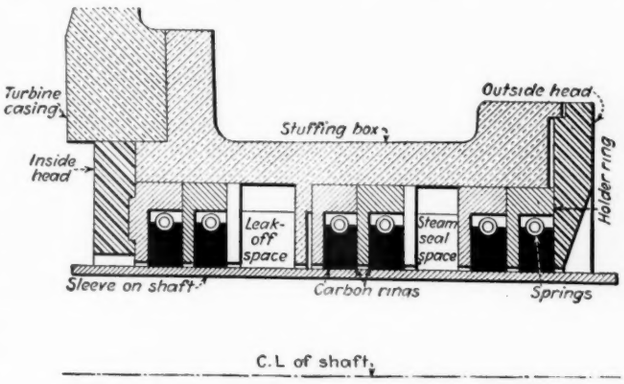


FIG. 2. CARBON-PACKING SHAFT GLAND

Conditions (*c*) and (*d*) occur when the turbine is being prepared for service after a shutdown. Fifteen to thirty minutes is customarily required to raise the vacuum in a large surface condenser to the point, about 20 in., where the turbine shaft may be easily turned over by steam. During this time a vacuum exists in all parts of the turbine shell, including the head and the steam passages between the head and the admission valves. This is condition (*d*). The next step in starting the turbine is to open the throttle valve enough to turn the shaft, giving a sudden admission of steam in volume. The throttle is then almost closed, just enough steam being admitted to keep the shaft turning at the slowest possible speed. The shaft and wheels and diaphragms are slowly warmed up in this manner for ten to twenty minutes, depending on the size of the turbine, after which the throttle is further opened and the machine slowly brought up to speed. A small vacuum will usually be noted during such a warming-up process and until load is picked up, in the first stage of the turbine, or the space marked *S* in Fig. 1, which is adjacent to the shaft at the head end of the turbine. Line (*c*) shows the pressures in the turbine under starting conditions. In either case, (*c*) or (*d*), there is a vacuum at both ends of the turbine shaft which may cause leakage of air inward with resulting decrease of the vacuum.

The problem of overcoming steam and air leakage at the head end and of air at the exhaust end has been solved in three ways—by carbon and labyrinth packing in connection with a steam seal, and by a water-sealed gland. In small turbines running non-condensing, soft packing is ordinarily used. The water seal is not applicable to the head end where the high steam pressures and temperatures would prevent its efficient operation, and when installed at the exhaust end, is usually supplemented by one of the other types of packing, for use in raising vacuum when the turbine shaft is standing. Carbon and labyrinth packing make no attempt to prevent leakage entirely, but by means of a steam seal substitute steam leakage for air leakage. Steam leaking in around the shaft has a negligible effect on the vacuum within, whereas a like volume of air, if admitted, would greatly impair it. At the head end the steam leakage outward is minimized by either packing, and such steam as escapes, is led by suitable connections to a region in the turbine where the pressure is similar to that of the escaping steam. That such a pressure stage exists in the turbine will be evident from the pressure lines in Fig. 1. The leakage at the head end may also be applied as sealing steam to the packing at the exhaust end of the shaft.

Carbon packing consists of a series of carbon rings built up of segments encircling the shaft and held against it by springs, so that any wear of the packing rings is immediately and evenly taken up. The carbon rings fit snugly in metal holder rings to minimize leakage over and around the packing rings. The rings are fitted in groups, as shown in Fig. 2, and the space

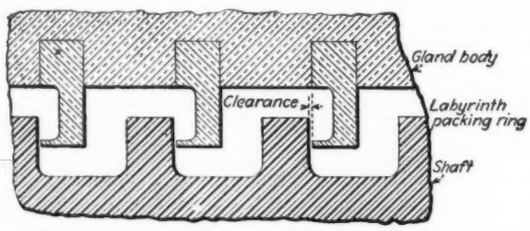


FIG. 3. LABYRINTH PACKING WITH AXIAL CLEARANCE

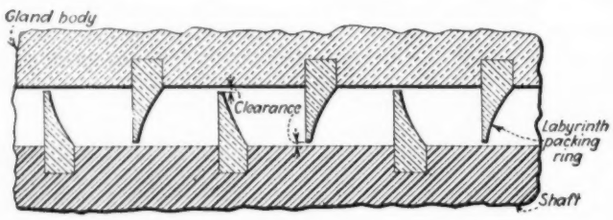


FIG. 4. LABYRINTH PACKING WITH RADIAL CLEARANCE

between groups used to collect the steam leaking by the rings from the interior of the turbine at the head end or to supply steam for sealing when installed at the exhaust end. Carbon packing, being able to adjust itself to any movement of the shaft, either axially, caused by excessive end thrust, or radially due to extreme vibration, is not injured as quickly thereby as labyrinth packing.

As its name suggests, labyrinth packing is designed to present a long and winding path to the flow of steam through the gland. The steam is throttled at each constriction in the path, with a reduction in pressure, so that starting with high pressure at the inside end of the gland, the steam escaping at the outer end will have only a few pounds and may be easily collected and piped elsewhere. If on the exhaust end of the turbine, steam of a slight pressure may be supplied at the outer end of the gland and escape through the packing into the turbine, being gradually reduced in pressure until the exhaust vacuum is reached. The packing is composed of rings alternately attached to the shaft and to the body of the gland. The clearance may be between rings, in the direction of the shaft, as shown in Fig. 3, or at right angles to the shaft, between tips of rings and the gland body and shaft, as illustrated in Fig. 4. It is evident that any movement of the turbine shaft relative to the shell may cause the packing rings to rub, so the clearance must be accurately determined and maintained. Soft metal, as copper or a composition, is used for the rings to prevent excessive heating in case the rings do rub. Drain-pipe connections are made to the bottom of the steam-collecting space to draw off any condensation that may occur in labyrinth packing, and similarly in carbon packing.

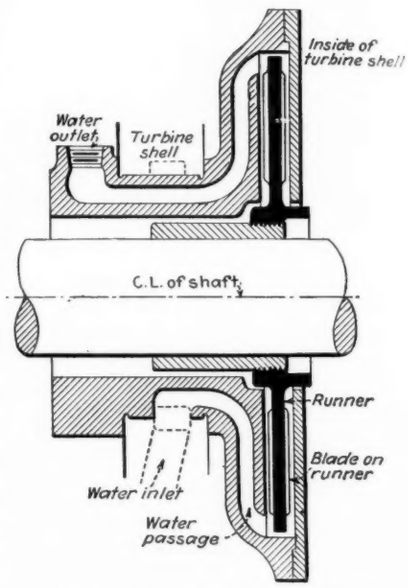


FIG. 5. TURBINE-SHAFT WATER GLAND

A water ring, or gland, can function only after the shaft has reached considerable speed, as it depends on the effect of centrifugal force for its formation. Fig. 5 shows a section of a water gland. A small runner, or set of blades, encircles the shaft and is inclosed around the rim and sides, with water passages to and from the space around the rim. Water under a moderate pressure, as 15 to 20 lb. per sq.in., is introduced in this space where it meets the revolving runner and is hurled outward, forming a revolving ring of water under pressure sufficient to resist the vacuum or exhaust pressure inside the turbine shell. The water may escape from the gland externally or flow into the turbine shell and thence to the condenser. A small flow must be maintained to keep the water from reaching a vaporizing temperature. Water containing any sediment or scale-forming salts must not be used for sealing water, as the heat and centrifugal action in the gland are liable to precipitate the solid matter and cause clogging.

Fig. 6 shows a design for a steam-water sealing system, embodying a steam seal and unloading valve at the head end of the turbine, and a steam seal and loading valve at the exhaust end; also a separate water seal at the low-pressure end. The loading valve is supplied with live steam and keeps steam on both seals when starting up, valves *M* and *N* being open and the unloading valve being closed. When load is on the turbine, valve *M* is closed, also *N*, and the unloading valve allows any excess leakage from the head-end packing to be

utilized in the turbine at *R*. The exhaust-end gland is sealed by turning on the water valve *W* after steam is shut off at *M*, provided the shaft has approached rated speed. If the water supply fails, or it is desired to use steam at the low-pressure packing when running, valve

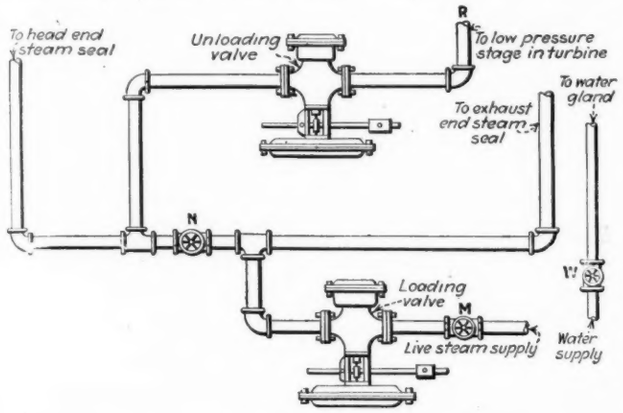


FIG. 6. STEAM-WATER SEALING SYSTEM

N is opened and leakage steam from the head-end packing is used at the exhaust end. Valve *N* may be open and the loading valve used to supply steam in case the pressure falls too low at the exhaust-end gland.

Store Refuse as Boiler Fuel

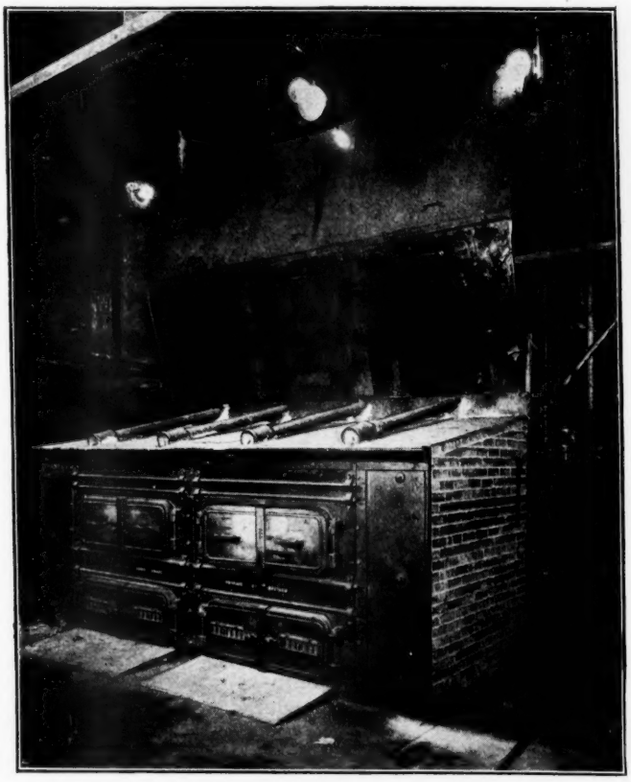
In a retail store or office building the disposal of rubbish in the form of waste paper, packing material, broken boxes, etc., is usually an item of expense and seldom an asset. The material cannot be burned to advantage in the ordinary boiler furnace, either in combination with coal or separately, so the usual plan is to provide an incinerator for the sole purpose of consuming it. This had been the practice in Rothchild's Department Store, Chicago. The incinerator was equipped with a plain grate, 3 x 5 ft., and required three men per day, one on each shift to fire it, and additional men to feed down the material from above. It had become too small for the service, and the danger of fire spreading to the material stored in the vicinity made new equipment or a change of methods essential.

In considering the problem, it was decided to abandon the old incinerator and to utilize the waste material as fuel in a new boiler installed for the purpose. The result has been a saving in coal and labor running from $35 to $45 per day during the short time the installation has been in service.

To supply steam to the compressor of a refrigerating plant for heating and to a number of restaurants, two water-tube boilers had been installed, each having 4,100 sq.ft. of evaporative surface and equipped with chain-grate stokers. The operating pressure averages about 140 lb., and although both units are required during the heating season, one can carry the load in summer. Additional heating load had been taken on recently, demanding more boiler capacity as reserve, and this was supplied by a new boiler having 4,000 sq.ft. of heating surface. It was not necessary to disturb the walls in any way, as the boiler is made up of sections that can pass through the usual doors or openings, and was taken down in the freight elevator to the boiler room in the third sub-basement.

Plenty of floor area was available crosswise of the setting, but lengthwise space was limited, and because of the piping already in place, a boiler requiring low headroom as well as small floor space was desirable. Both these features were secured in the new boiler, which occupies a floor area of 196 sq.ft., or 0.49 sq.ft. per 10 sq.ft. of steam-making surface, and measures 18 ft. 7 in. from the floor line to the steam outlet, allowing a dutch oven with an extension of 5 ft. 6 in. from the boiler front and room for rubbish in front of the furnace. The material slides down a chute from above to a convenient location for the fireman to shove it into the furnace.

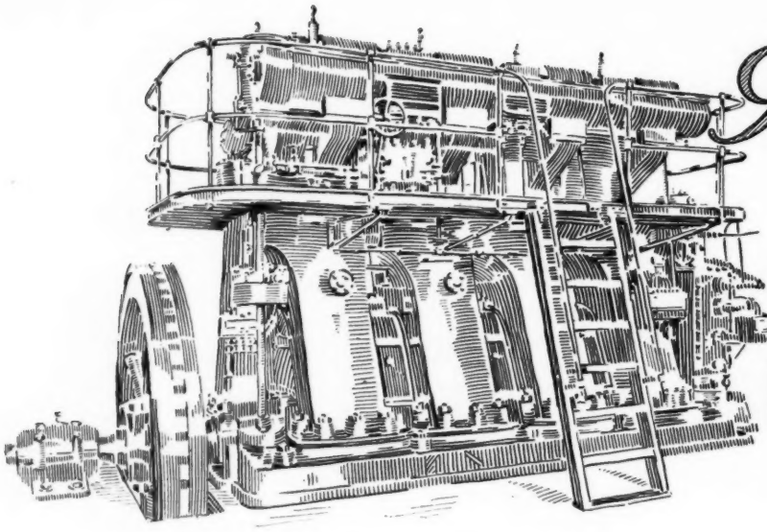
It was recognized that some coal would be necessary to fill in the intervals in the supply of the waste material and to supplement this fuel when the quantity available was running light. To avoid the difficulties involved in burning the two fuels on one grate, the boiler was equipped with a down-draft furnace having 60 sq.ft. of surface on the upper grate and 60 sq.ft. on the lower. The latter is nearly on a level with the



REFUSE-BURNING BOILER IN DEPARTMENT STORE

boiler-room floor, so that the waste material is readily shoved onto it through large doors, and when burned, the ash may be dumped into a large pit below the furnace.

Coal is hand-fired onto the upper grate, and the fuel bed is maintained in condition to respond in the light intervals of waste supply or assume the entire load when the quantity available has been depleted. The combination has worked out nicely. Full rated capacity can be maintained on the boiler and smokeless operation is obtained. During the summer months this unit is expected to carry the entire load of the plant. Two men have been eliminated from the crew formerly handling the waste material, and as less coal is burned, these two items make up the daily saving previously indicated.



Indicating the DIESEL ENGINE

By J. A. Walland

THE Diesel engine is a high-grade machine, and to obtain results it must have good care. By results is meant, first, reliability, and second, economy. These two requirements are closely tied together because proper combustion means economy of fuel and moderate wear and tear on the engine as a whole, which in turn helps tremendously toward reliability. An indicator is as important in studying and maintaining good combustion as a pressure gage is in indicating spray-air pressure, or a thermometer is in showing the cooling water or exhaust temperature.

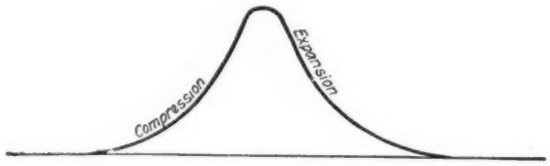


FIG. 1. COMPRESSION DIAGRAM

The operator need not think that the indicator is a laboratory or test-stand adjunct only. It is a very practical instrument for frequent use in the power station and really does not require extraordinary skill for its use. It should of course be treated with respect, but given a good instrument to start with, it only asks to be oiled each time before use, a careful cleaning after use, together with careful handling, in order to be of real assistance.

When buying an indicator, select only a type made especially for Diesel engines by a reputable firm. The instrument must be more substantially built throughout and higher-grade materials used than is the case with steam-engine indicators. The best indicator to be had is not too expensive. As already intimated, combustion is the most important event in the whole cycle of operations. The inlet and exhaust strokes on a four-stroke-cycle engine are taken care of by proper timing of the inlet and exhaust valves, and this timing is most readily checked with the engine stopped. Even if the timing is several degrees off from standard, no harm will be done. The same applies to the exhaust and scavenging events on a two-stroke-cycle engine. But it is vital to have the proper compression pressure and combustion. The compression and working, or expansion, strokes will then be satisfactory. Consequently, the part of the entire cycle of operations that the operator must study closely is confined to a relatively small crank angle in the vicinity of top dead center on the working stroke. Even the closing of the fuel-injection valve can be slighted and is relatively unimportant, but the opening of the valve demands the operator's closest attention.

The indicator is the one instrument that tells the whole story of these important events, but with the ordinary form of closed indicator diagram they are so crowded together at one end that it is always difficult and oftentimes impossible to read the story. To be sure, the closed diagram tells the indicated horsepower, but ordinarily this need not worry the operator. In the case of multi-cylinder engines with a single fuel pump for all cylinders, this of course is about the only means of balancing the power between the cylinders. Most engines, however, have individual pumps for all cylinders, and the cylinders are readily and very closely balanced by mechanical adjustment of the pump. The obvious thing to do, then, is to take the kind of indicator diagram that will best show the events at the end of the stroke. On a steam engine we would take an "eccentric diagram," but on an oil engine the corresponding method is to rotate the drum with the crank at the instant of passing top dead center. It is not necessary to do this mechanically, because it is not at all vital that the rate of rotation of the drum should correspond exactly with that of the crank. Simply attach the indicator to the cylinder as usual, turn on the cock and pull the indicator drum string by hand quickly. It takes very little practice to be able to synchronize the pulling of the indicator drum string with the time of combustion in the engine. Then you will have an indicator diagram with the compression and combustion events

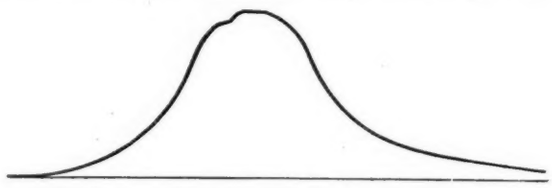


FIG. 2. NORMAL FUEL-VALVE TIMING

spread over considerable space instead of jammed together at one end. Incidentally, this kind of diagram is simpler and quicker to take because you do not have to fuss with hooking up and unhooking to the piston motion.

The next thing is to study these diagrams and be able to read what they tell. Right here let it be suggested that a scale of 400 lb. per inch be used in order to make the card easy to read. This usually means a piston of one-quarter square inch area and a 100-lb. spring.

The first thing to study is the compression pressure, for upon this depends all else. A straight compression diagram with fuel cut off the cylinder is shown in Fig. 1. This tells the story of the tightness of valves and piston rings, and the correctness of the clearance volume after the replacing of any cylinder heads, pistons, or adjustments of bearings. Leaks always cause a reduction in the height of the diagram, but a change in clearance volume may increase or decrease the height. Aside from an examination of the valves themselves a leaky inlet or starting valve can generally be detected by a hissing sound and a warming of the pipes adjacent to the valves, a leaky exhaust valve by a hissing sound and a darkening of the exhaust. Leaky piston rings will, of course, permit blowing by the piston. Leaky spray valves produce opposite results from leaky inlet or exhaust valves. The pressure is raised in this case, and generally the cylinder relief valves are lifted. When taking a straight compression diagram, it is not absolutely necessary to shut off the injection air, although

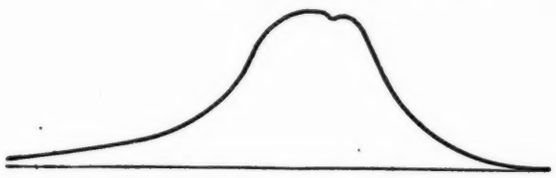


FIG. 3. LATE FUEL-VALVE TIMING

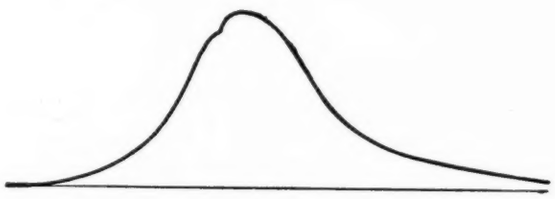


FIG. 4. EARLY FUEL-VALVE TIMING

it is better to do so in doubtful cases. In several cases known to the writer, the compression pressure was taken under both conditions, but absolutely no difference could be detected.

The next thing is the timing. Normal timing is shown in Fig. 2, late timing in Fig. 3 and early timing in Figs. 4 and 5. Figs. 3 and 4 do not show extreme cases, but give an indication of the usual appearance of the diagram under the conditions named. Note that in the perfect diagram the pressure increases along the compression curve to the maximum or compression pressure, hesitates for an instant, and then merges into the ignition, combustion and expansion lines. The compression line must become horizontal at the top and yet it must not start to droop for perfect timing. This means, then, that ignition takes place just as the piston is passing dead center. The proper procedure is to set the injection valves as nearly right as possible with the engine stopped, and correct them to a nicety by taking cards with the engine in operation. An extreme case of early timing will often merge the ignition and combustion lines into the compression line in such a way as to make the diagram look similar to a straight compression diagram, and then the compression pressure will look perfectly normal when actually it may be very low. This is an extreme case, however, and is

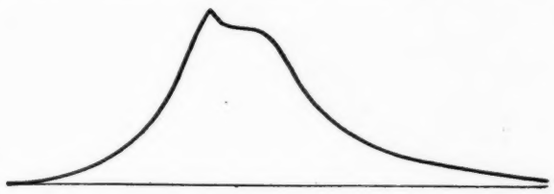


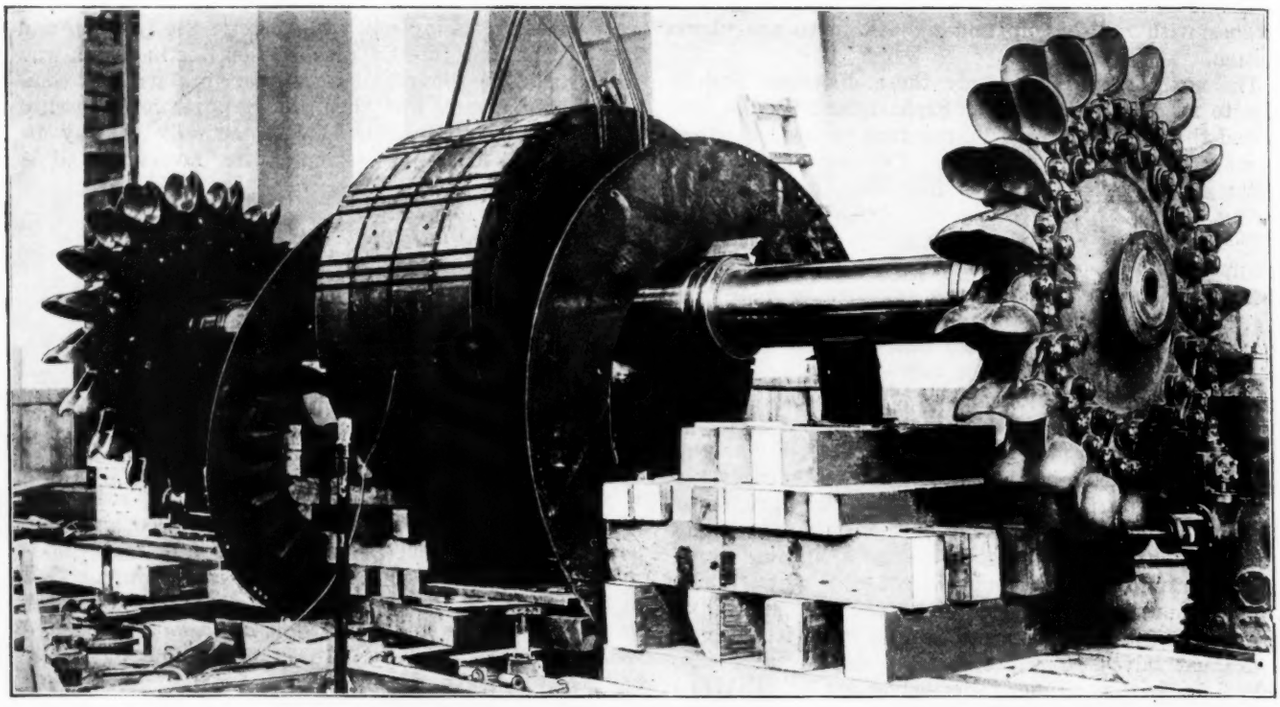
FIG. 5. EXTREME CASE OF EARLY TIMING

rarely met with in actual practice. Throughout all this it must not be overlooked that a wild injection air pressure may have an unfortunate influence and give a misleading diagram. An extremely low or high pressure may make the diagram look like late timing. This again is not a likely occurrence since the operator knows, or ought to know, the right injection air pressure within very close limits. It is proper and desirable, however, to make minor changes in this pressure to assist in obtaining perfect combustion conditions, and this brings us to the third subject in connection with indicator cards which ought to be studied.

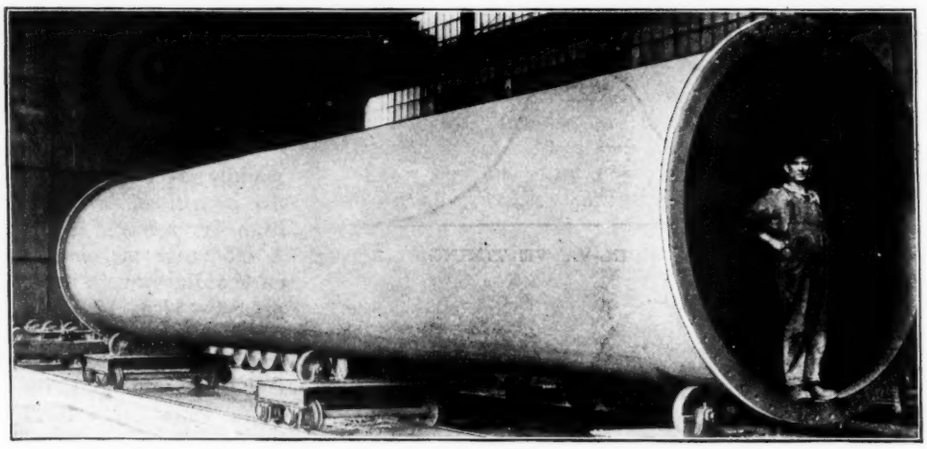
This subject is the shape of the combustion line. The ideal shape is, of course, the perfectly flat top with a pressure equal to, or only slightly greater than, the compression pressure. If the pressure falls away rapidly after ignition, try a little higher injection pressure or a little earlier timing. If the pressure rises too high, try a lower injection pressure and a later timing. A moderate increase in pressure and a rounding over on the diagram is not at all undesirable. But when the pressure rises 100 lb. and more above the compression pressure, it is time to do something.

While the foregoing remarks apply particularly to full load conditions, the same principles apply at fractional loads. The level portion of the diagram at the top is naturally not so long under these conditions, but everything else is just the same. The length of the level part is not exactly proportional to the load, owing to the effect of the injection air.

The main idea is that the taking of the kind of indicator diagrams described herein is so little trouble that the operator has a positive incentive for doing it. Given these diagrams, the information to be obtained is much more complete and valuable than what can be obtained from the regular closed diagrams; the study is fascinating, and the result is an increased knowledge and understanding of what takes place in the cylinder, to say nothing of the gain in smoothness and reliability of operation. If the operator thinks that this last statement is a little overdrawn, let him pause to consider the damage that imperfect combustion can do to the valves, piston and cylinder, and what increased duty is put upon the bearings by excessive cylinder pressures. It requires far less effort in the long run to keep an engine adjusted and running properly than to neglect the adjustments until something goes wrong and then make repairs.



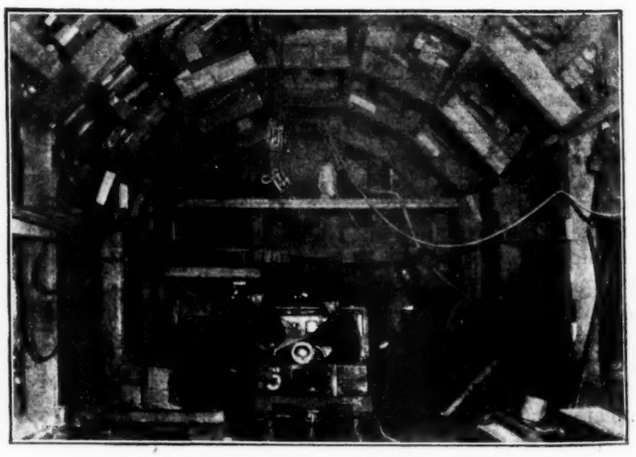
Revolving element of a 20,000-hp. overhanging-shaft twin-impulse turbine for the Pacific Light and Power Co.'s No. 1 Plant. It was built by the Allis-Chalmers Co. for a head of 1,900-ft. and a speed of 375 r.p.m.



A 96-in. welded steel pipe which is to carry the exhaust from a 10,000-kw. turbine to a barometric condenser. It is made of rolled steel plates forge-welded by the water-gas process and equipped with cast-steel flanges welded in a similar manner. The pipe is 43 ft. 6 in. long, and the walls are ¾ in. thick. It was built for the Bethlehem Steel plant at Lebanon, Pa., by the American Welding Co.

One seldom thinks of a desert in connection with water power, yet here are both close together. This concrete flume carries water for the Arizona Power Co. through the hills at Fossil Creek.
Courtesy of the National Electric Light Association.

Below, the 8,300-ft. tunnel for the Kern Canyon Power plant of the San Joaquin Light and Power Corp. being enlarged. For about 2,000 ft. it is to have a cross-section of 100 sq.ft., lined with concrete; the remainder will be about 13x14 ft.



Record System of Steam and Motor Pumping Costs*

By O. A. ANDERSON
Motive Power Department, Armour and Company, Chicago, Ill.

THE pump report functions as a clearing house for all pumping expense that is incurred in all branches of the power-house department. These sheets form a record of the quantity of water or brine pumped for each individual or group of pumps in such a manner that the respective charges may be distributed to the proper branch of the motive-power service. The cost of operating boiler feed-water pumps is charged to the cost of producing steam and carried forward to its proper place in the motive-power reports. Pumps handling water over the ammonia and steam condensers are recharged to refrigeration and steam condensing engines, respectively. Brine-pumping expense is also recharged to refrigeration. The major part of the pumping expense, however, is charged to water service compiled as in Table II.

Description of the pump and details pertaining to the location of the suction and discharge connections are always filled in under the proper headings of item 1, Table I. Items 1(g) to 1(j) are obtained from the power-house daily record, average and total values being used for the period over which the report is made. By referring to items 2(a) to 2(d) inclusive, it will be apparent that the form has been made up primarily for reciprocating pumps. In the case of centrifugal pumps, however, the gallons per minute capacity values may be substituted in item 2(c) and the total gallons delivered by multiplying the gallons per minute by the number of hours of pump operation, all multiplied by sixty in order to convert to total minutes of operation.

Record system to determine cost of pumping by motor or steam-driven pumps or by the air lift for all services. Useful tables are included on the quantities of water required for condensers of steam and refrigerating systems and the amount of air per gallon of water in the air-lift system.

In item 2(e) the hydraulic horsepower-hours represented is determined by the use of the formula,

$$Total\ hp.\text{-}hr. = 0.00972 \times av.\ hd.\ press.\ in\ lbs. \times total\ M\ gal.\ pumped$$

To simplify the calculation of steam consumption of the different pumps, Table III has been prepared, showing the values of steam consumption per hydraulic horsepower, for different types of pump and prime mover.

With the proper steam rate, the total steam consumption is obtained by multiplying item 2(e) by 2(f). According to whether the exhaust steam is used or discharged to the atmosphere, one-third or full charge is made for the steam figures calculated and the value indicated in item 2(i). The cost of steam is determined for each pump or group of pumps by multiplying the value of 2(i) by the unit cost of the steam, the resulting value being filled in opposite item 2(j).

Items 2(k) to 2(m) have been prepared for motor-driven pumps. A value of 1.25 kw. per h. hp. (hydraulic horsepower) is used for motor-driven reciprocating pumps and 1.35 kw. for motor-driven centrifugal pumps.

Items 3, 4 and 5 cover labor expense, supplies and repairs which are chargeable direct to pump operation. In a large plant frequently a separate pumphouse is in operation, containing a separate set of engineers and operators. In these instances the matter of keeping such a cost record is greatly simplified. In smaller plants estimates on the division of the whole power-house expense will have to be made to apply to the pump operation. Items 6 and 7 are the proportionate charges for steam condensing and power-house lighting.

After rental and depreciation values are added to the costs of operation, the total cost of pumping is divided by the thousands of gallons pumped and the cost per thousands gallons reported in item 10(b) of Table I. Pump report No. 2 is an extension of Table I. In most plants it is necessary to use several such blanks in order to serve all the pumps. Pump calculation is carried through on the basis of data included under item 1.

As will be seen by referring to Table II, there are two major divisions; namely, purchased water and local water. By purchased water is designated such water as is purchased through meter from a utility corporation or a municipality. Local water is that pumped from wells or rivers by means of the plant's own pumping equipment. This branch of service supplies water to all the departments of the factory and fills the requirements of the power house. All the expense incurred is recorded on this sheet according to the proper divisions. This tabulation is not designed to absorb pumping expense in connection with recirculating water over cooling towers and ammonia condensers, but is intended to carry the expense of producing and delivering fresh water to the requirements of the plant. Water recirculating expense is charged to either ammonia or steam condensers.

Item 1 is obtained from meter readings and item 3 determined from the average water expense rate filled in as item 2. Item 4 is the summation of two items 2(i) of Table I for pumps whose operation is chargeable to water service. Item 5 is obtained in the same manner except for motor-driven pumps. The cost for steam and electric power is also obtained from Table I for the water-service pumps and shown as items 6 and 7 of Table II. Likewise, costs of labor, supplies, repairs, rent and depreciation are transferred from the pump report to item 8. A summation of items 6, 7, and 8 will give the total pumping expense shown as item 9. The total cost of purchased water will be the sum of items 3 and 9, indicated as item 10. The cost per thousand gallons is obtained by dividing the total cost in item 10 by the thousands of gallons in item 1. Items 12 to 21 are designed to cover pumping expense for water obtained from wells. These items include, however, the total pumping cost to the factory departments or power house as distinguished from the cost of pumping the water only to the surface of the well, as shown in Table IV.

The total thousands of gallons of water from wells is obtained from item 11 of Table IV and properly filled

*Articles by the author on the system of keeping records or other equipment at the Armour plant appeared Jan. 25, Feb. 1 and Feb. 8, 1921.

in as item 12 of Table II. The total thousands of cubic feet of free air used for pumping water with air is obtained from item 18 of Table IV. In the case of steam pumps operating in wells, the figure is taken from item 36, Table IV, and for power pumps from item 15. Costs for air, steam and electric power are indicated in items 16, 17, and 18 of Table II.

To the pumping expense from the wells there must be added the steam and electric power expense required to pump this water to the respective departments under pressure. This information is obtained from Table I. Reference will be made to the proper pump and service. Similarly, the cost of labor, supplies, repairs, rent, etc., is transferred to Table II.

Items 22 to 31 are designed to cover the operation of pumping equipment handling water from rivers or other sources than wells where no pumping privileges are involved. All the information required in these items will be obtained from the summation of the proper values for the pumps performing duty on this kind of water service, as shown in Table I. The purpose of distinguishing between different methods of water service is to have shown at all times the comparative economy of one service with the other.

Operation of Wells

The "Well Report" forms are work sheets on which is developed the cost of pumping water from different levels in the wells to the surface reservoirs. The summarized information is referred to Table II, which is the main water report. Examination of items 1 to 11 of Table IV will show that these items call for a description of the wells and a record of hours of service during the period and capacities of each well in gallons per minute. Items 6 and 7 indicate water levels when the well is down and when it is in operation.

When the depths of wells are 300 ft. and under, levels are usually obtained by dropping a small-sized air pipe with the end submerged and air pressure applied. Reading of the pressure gage is noted at the point where it will not increase. This is the point of pressure balance between the air in the pipe and the water in the well above the bottom of the pipe. The water level is then obtained by subtracting from the length of the air pipe the air pressure in pounds per square inch, multiplied by 2.31. In wells over 300 ft. deep the depth is obtained usually by sounding with a cord and wooden float or by the short-circuiting of one pole of an electric circuit of weighted wire with water surface.

Items 12 to 19 pertain only to wells where the air lift is employed. Air is the prime mover, and data leading to the quantity of air required are formulated. The manufacturers of air compressors and air-pumping engineers have made extensive tests indicating that the consumption of air depends largely upon the depth of submergence of the air-supply pipe. Accordingly, the percentage of submergence is determined, to obtain from Table V the quantity of free air required to lift one gallon of water to the surface.

Percentages of submergence are figured for the conditions existing when the well is down and when in operation. The value applied to the air-lift table is that for the submergence when pumping. It is obtained by subtracting the depth of the water level when pumping from the total depth of the air pipe from the point of the air discharge in the well to the water discharge at the surface and dividing by the last-mentioned figure; namely, depth of air pipe. This fraction is converted to a percentage value by multiplying by 100. As an example refer to a well of 1,000 gal. per min. capacity drilled to a depth of 300 ft. and having a water level of 70 ft. when pumping at capacity. The depth of the air-supply pipe is 200 ft. The percentage of submergence is obtained as follows:

$$\textit{Per cent submergence} = 100 \times \frac{200 - 70}{200} = 65$$

Referring to Table V, read down the vertical column under the heading "Lift in Feet" to 80 and then read across horizontally to the column headed 65 per cent, and it will be noticed that the submergence is shown as 149 ft. Following the 60-ft. lift horizontally across, there will be a submergence of 111 ft. Interpolating between these two values, it will be found for a lift of 70 ft. that the submergence will be 130 ft. and there will be required 0.343 cu.ft. of free air per gallon of water pumped.

After the number of cubic feet of air per gallon is obtained, the total air used for the period may be determined by multiplying this value by the number of gallons pumped for the period, and the cost by referring to the unit cost of compressed air given in a previous article.[1] Where an individual or group of air compressors operate on well pumping only, it will be, of course, more accurate and simple to use directly the total quantity of air pumped, when establishing the cost.

It will be noticed that items 20 to 27 of Table IV pertain to motor-driven pumps and items 28 to 40 to steam-driven pumps. These items apply particularly to centrifugal pumps, which are rapidly coming into universal use for large capacities. At the time of acceptance of new pumping equipment, tests are made usually on the capacity by means of a weir box, the efficiency being obtained from electric-meter readings in the case of electric motors and condensation tests or the turbines. These test data are used in compiling the cost of operation on the report and are verified at different periods throughout the life of the equipment.

In larger plants there is at least one man whose duty it is to care for the operation of the well-pumping and and water-service equipment. This labor expense, together with supplies, repairs and overhead of the well equipment, are chargeable to this service and are indicated in items 42, 43 and 44. The summation of the well-pumping expense is shown in item 45(a), which is referred to Table II, according to the proper classification of expense.

Well water is extremely desirable at a packing plant, owing to the large quantities of cold water required for ammonia condensing in connection with the refrigeration machinery. With the low temperature of the well water less is required and the head pressure on the ammonia condensers is reduced, resulting in a saving of fuel. Further, with the large quantities of water required for this purpose, the cost of pumping will be lower for a centralized pumping plant, as compared to water from utility or municipal pumping plants.

Steam and Ammonia Condensing

A description of the arrangement of connections and water service to the ammonia and steam condensers is reported in items 1 to 5 of Table VI. Usually, the ammonia condensers are located at a level above the steam condensers so that the water discharges by gravity from one to the other. The time is approaching when a steam condenser will be a thing of the past in

[1] "Armour System of Power Plant Accounting," Jan. 25, 1921.

a packing plant, as there are so many uses for exhaust steam.

When the steam condensers are in service, the total steam condensed for the compressor engines, electric generating units, etc., is referred to item 6 and properly tabulated so that the total may be obtained and the proper percentage of the total determined in order to be able to distribute the expense of pumping in connection with the condenser service. The total steam in thousands of pounds is given in item 7.

To estimate the water required per pound of steam under variable conditions, Table VII has been prepared. It has been calculated for a 22-in. vacuum, but will not be in error over 1 per cent if used between 12 and 27 in. of vacuum. The value obtained from the table is filled in as item 8 of Table VI. Multiplying by item 7, there will result the total thousands of gallons of water required and reported as item 9.

When the water passes in series over cooling towers, ammonia condensers and steam condensers, items 10 to 21 will be figured, but if the water passes separately over the ammonia and the steam condensers, the calculations will be made for items 22 to 31 inclusive. The total thousands of gallons pumped as reported in item 10(b) may be determined by summing up the total gallons of water handled by the pumps circulating water over the cooling tower, ammonia condensers and steam condensers. Items 11 and 12 indicate that a charge is made for pumping in addition to the pumping expense. This charge is for makeup water only to replace losses over the cooling tower and condensers. The pumping and labor expense will be obtained from the pump sheets for the pumps operating on this particular service. The total cost of operating the condensers is usually divided 50 per cent to ammonia condensing and 50 per cent to steam condensing; otherwise careful figures may be prepared on the basis of the different heights of the ammonia and steam condensers above the pump.

As before mentioned, items 22 to 31 show the pumping operations divided. The quantity of water handled and the pumping expense is obtained separately for each service, according to the number of pumps operating, as in Table I. As a check on the quantity of water passing over the ammonia condensers, Table VIII has been prepared. A charge for the water is made only for makeup or when all the water discharges to the sewer, as will be the case in a plant where the repumping expense plus the overhead on the cooling tower will equal or be greater than the cost of pumping the cold water from the wells.

Items 32 to 47 in Table VI represent work-sheet figures for developing the loss of water over the ammonia condensers and cooling towers. These figures are usually not prepared except in the case of making an analysis of loss on a special investigation. Usually, a test or estimate will suffice.

Items 48 to 53 include the summarized expense taken from Table I for the vacuum pumps. This expense is of course chargeable to steam condensing and is included with the balance of the steam condensing costs to make up the total shown in item 58. The ammonia-condenser costs are summarized in items 20 and 31 and referred to the motive-power report sheet on refrigeration. The steam-condenser costs, however, are divided between refrigeration, power and light, air compression and water pumping, according to the quantity of steam condensed, as given by percentage values in item 6 of Table VI.

Table IX is a recapitulation of all of the total power-house production units and costs. The total costs for steam, refrigeration, etc., are all transferred to the top line as "Total Net Produced." It can be seen that the summation of all these items will not represent the true total cost of operating the power house, as the cost of steam has been charged under the heading of steam for refrigeration, power and light and pumping, and still the total steam expense was included in all the charges in the motive-power report sheets presented in the first article. It will be seen that proper deductions must be made from the total energy produced in each branch for that part of the same energy which is used in the power house. For instance, under the heading of "Steam" the steam units and expense of the refrigeration division are deducted and the same thing is done with the steam used by the other branches.

Referring to the figures on the bottom line of Table IX, showing the net units of service to the factory and their costs, it will be apparent that the summation of the costs will be equal to the net balance shown by the accounting-department ledger chargeable to the power-house service. Percentage values for cost of steam, refrigeration, etc., are usually prepared from the costs shown on this line and applied to the accounting department's expense value and each department of the factory charged for this service according to the quantity of power-house service rendered during the period.

In closing, the writer would say that this system of records is functioning successfully at a number of large plants, and in the smaller plants the same system is used except in a more abbreviated form. For the material presented thanks are due to P. W. Evans, motive-power department of Armour & Co., who originated and developed the system of reports under discussion. A. McKenzie is in charge of the motive-power department and has supervision of all Armour power plants.

TABLE I. MOTIVE POWER PUMP REPORT

1. Data on operation of pumps:				
(a)	Pump No.	4	6	7
(b)	Type of pump	Duplex	.....	.....
(c)	Type of drive	.....	.....	.....
(d)	Suction taken from	Heater	Brine	Brine Cooler
(e)	Discharge to	Boilers	Whse.	Hend. Cooler
(f)	Operation chargeable to	Boilers	Brine	Brine
(g)	Total hours run	1,176	657	1,344
(h)	Avg. speed, r.p.m.	22.5	20	36
(i)	Total revolutions made	1,587,600	788,400	2,903,040
(j)	Avg. head in lb.	128	28	36
2. Calculation of delivery and power required:				
(a)	Assumed per cent slip	15	15	15
(b)	Gal. per rev. total disp.	7.78	23.27	23.02
(c)	Gal. per rev., less slip	6.61	19.78	19.55
(d)	Total M. gal. delivered	10,494	15,595	56,754
(e)	Total h.hp.-hr.	13,056	4,244	19,860
(f)	B.hp. per h.hp.	4	4	4
(g)	Total B.hp.-hr. to pump	52,224	16,976	79,440
(h)	Exhaust steam to	Heater	Heater	Heater
(i)	B.hp.-hr. consumed (½ to al if exhaust is used)	17,408	5,659	26,480
(j)	Total cost steam	.....	.....	.....
(k)	Kw.-hr. per h.hp.	.....	.....	.....
(l)	Total kw.-hr. consumed	.....	.....	.....
(m)	Total cost electric power	.....	.....	.....
3. Cost of labor for operation of pumps:				
(a)	Portion of engineers' wages	$.....	.....	.....
(b)	Pump tenders	$.....	.....	.....
(c)	Janitor service	$.....	.....	.....
(d)	Testing	$.....	.....	.....
(e)	Reports	$.....	.....	.....
(f)	.....	$.....	.....	.....
(g)	Total cost of labor	$.....	.....	.....
4. Cost of supplies:				
(a)	Lubricants:			
	——gal. of—— ——@ $——per gal..	$.....	.....	.....
	——gal. of—— ——@ $——per gal..	$.....	.....	.....
	——gal. of—— ——@ $——per gal...	$.....	.....	.....
	——lb. of—— ——@ $——per lb.	$.....	.....	.....
(b)	Waste	$.....	.....	.....
(c)	Packing	$.....	.....	.....
(d)	.....	$.....	.....	.....
(e)	.....	$.....	.....	.....
(f)	Total cost supplies	$.....	.....	.....

TABLE I, CONTINUED. MOTIVE POWER PUMP REPORT

5. Cost of repairs:
   (a) Pumps ..... $
   (b) Motor driving pumps ..... $
   (c) Turbines driving pumps ..... $
   (d) Piping ..... $
   (e) Total cost repairs ..... $
6. Portion of total steam condensing cost ..... $
7. Electric lighting for pumps ..... $
   ——— kw.-hr. @ $—— per kw.-hr. ..... $
8. Total cost pumping equip. and connections ..... $
9. (a) Annual rental charge, 6% ..... $
   (b) Annual depreciation charge, 5% ..... $
   (c) Rental charge for period ..... $
   (d) Depreciation charge for period ..... $
10. (a) Total cost of pumping ..... $
    (b) Pumping cost per M. g. l. ..... $

TABLE II. MOTIVE POWER WATER REPORT

Purchased Water

1. Total M. gal. purchased	14,694
2. Cost per M. gal.	
3. Total purchase cost water	
4. B.hp.-hr. consumed in pumping	4,688
5. Kw.-hr. consumed in pumping	
6. Total cost steam for pumping	
7. Total cost electric power for pumping	
8. Total cost labor, supplies, repairs, rent, etc., for pumping	
9. Total cost for pumping	
10. Total cost for water and pumping	
11. Total cost per M. gal.	

Local Water

12. Total M. gal. from wells	
13. M. cu.ft. free air for pumping	
14. B.hp.-hr. consumed in pumping	
15. Kw.-hr. consumed in pumping	
16. Total cost compressed air for pumping	
17. Total cost steam for pumping	
18. Total cost electric power for pumping	
19. Total cost labor, supplies, repairs, rent, etc., for wells	
20. Total cost water from wells	
21. Total cost per M. gal.	
22. Total M. gal. local water from source other than wells	96.120
23. Water obtained from	River
24. Water delivered to	Steam cond.
25. B.hp.-hr. consumed in pumping	
26. Kw.-hr. consumed in pumping	44,411
27. Total cost steam for pumping	
28. Total cost electric power for pumping	$1,772.44
29. Total cost labor, supplies, repairs, rent, etc., for pumping	$50.709
30. Total cost water	2,279.53
31. Total cost per M. gal.	0.0237

TABLE III. STEAM CONSUMPTION OF PUMPS

Type of Pump	B. Hp. per H. Hp.
Direct-acting simplex pump	3.5
Direct-acting duplex pump	4.0
Direct-acting compound pump	2.5
Simple non-condensing Corliss pump	0.8
Simple non-condensing Meyer valve pump	1.0
Compound non-condensing Corliss pump	0.67
Compound non-condensing Meyer valve pump	0.75
Compound condensing Corliss pump	0.45
Compound condensing Meyer valve pump	0.55
Turbine, non-condensing	2.00

TABLE IV. OPERATION OF WELLS

1. Well number	1			
2. Method of pumping	Air lift			
3. Depth of well, feet	300			
4. Diameter at bottom, in.				
5. Diameter at top, in.				
6. Water level—well down	25			
7. Water level—well in operation	70			
Depth of air pipe	200 ft.			

TABLE IV. CONTINUED. OPERATION OF WELLS

8. Water delivered to				
9. Total hours in service				
10. Average gal. per min.	1,000			
11. Total M. gal. delivered				

Air Lifts

12. Per cent submergence, well down				
13. Per cent submergence, well in operation	65			
14. Air pressure—to blow off				
15. Air pressure—to operate				
16. Cu.ft. free air per gal.	0.343			
17. Total M. cu.ft. free air used				
18. Cost per M. cu.ft. free air				
19. Total cost compressed air				

Motor-Driven Pumps

20. Type of pump				
21. Make				
22. Speed				
23. Hp. of motor				
24. Actual kw. required				
25. Total kw.-hr. used				
26. Cost per kw.-hr.				
27. Total cost electric power				

Steam-Driven Pumps

28. Type of pump				
29. Drive				
30. Make				
31. Speed				
32. Steam consumption, lb. per in. hp.-hr.				
33. Steam consumption, lb. per h.hp.-hr.				
34. B.hp. per h.hp.				
35. Total hydraulic hp.-hr.				
36. Total boiler hp.-hr.				
37. Exhaust steam to				
38. Boiler hp.-hr. consumed				
39. Cost per b.hp.-hr.				
40. Total cost steam				

All Wells

41. Labor				
42. Supplies:				
(a) Lubricants,				
——gal. of—— ——@ $——per gal.				
——gal. of—— ——@ $——per gal.				
——lb. of—— ——@ $——per lb.				
(b) Waste				
(c) Packing				
(d)				
(e)				
(f)				
(g) Total				
43. Repairs:				
(a) To well				
(b) To water piping above ground				
(c) To steam and air piping to header or main line				
(d) To pumps				
(e) To motors driving pumps				
(f) To turbines or other steam drives				
(g)				
(h)				
(i) Total cost repairs				
44. (a) Total value of well complete with means for pumping				
(b) Annual rental charge, 6%				
(c) Annual depreciation charge, 5%				
(d) Rental charge for period				
(e) Depreciation charge for period				
(f) Insurance				
(g) Taxes				
45. (a) Total cost air, electric power, steam, labor, supplies, repairs, rent				
(b) Total cost per M. gal. delivered				

TABLE V. AIR LIFT TABLE—CUBIC FEET FREE AIR ACTUAL PER GALLON AND SUBMERGENCE IN FEET

Ratio of Submergence to Lift, per cent

Const., Formula C	Lift in Feet	33 Cu. Ft. Free Air	33 Sub. in Feet	35 Cu. Ft. Free Air	35 Sub. in Feet	40 Cu. Ft. Free Air	40 Sub. in Feet	45 Cu. Ft. Free Air	45 Sub. in Feet	50 Cu. Ft. Free Air	50 Sub. in Feet	55 Cu. Ft. Free Air	55 Sub. in Feet	60 Cu. Ft. Free Air	60 Sub. in Feet	65 Cu. Ft. Free Air	65 Sub. in Feet	70 Cu. Ft. Free Air	70 Sub. in Feet
246	20	....	....	....	....	....	....	....	....	0.32	20	0.27	25	0.24	30	0.20	37	0.18	45
245	30	....	....	....	....	....	....	0.41	25	0.36	30	0.31	37	0.27	45	0.23	56	0.20	70
244	40	....	....	....	....	....	....	0.43	33	0.41	40	0.34	49	0.30	60	0.26	74	0.23	93
243	50	....	....	....	....	....	....	0.48	41	0.42	50	0.37	61	0.325	75	0.29	93	0.255	117
242	60	....	....	....	....	0.60	40	0.52	49	0.45	60	0.40	73	0.35	90	0.32	111	0.28	140
240	80	....	....	....	....	0.654	53	0.57	66	0.528	80	0.453	98	0.41	120	0.367	149	0.32	187
238	100	....	....	....	....	0.71	66	0.64	82	0.564	100	0.508	122	0.46	150	0.415	186	....	....
235.5	125	....	....	....	....	0.79	84	0.71	102	0.63	125	0.58	153	0.52	188	0.48	232	....	....
233	150	....	....	....	....	0.93	100	0.795	123	0.702	150	0.64	184	0.586	225	....	....	....	....
230.5	175	....	....	1.06	94	0.95	117	0.85	143	0.77	175	0.70	214	0.647	262	....	....	....	....
228	200	....	....	1.13	108	1.02	133	0.92	164	0.84	200	0.77	245	....	....	....	....	....	....
223	250	....	....	1.28	135	1.17	167	1.07	205	0.973	250	0.90	306	....	....	....	....	....	....
218	300	....	....	1.45	162	1.31	200	1.21	246	1.12	300	....	....	....	....	....	....	....	....
213	350	....	....	1.62	189	1.47	233	1.38	286	1.26	350	....	....	....	....	....	....	....	....
208	400	....	....	1.78	215	1.63	267	1.51	327	1.39	400	....	....	....	....	....	....	....	....
203	450	2.03	222	1.95	242	1.79	300	1.66	368	1.54	450	....	....	....	....	....	....	....	....
192	500	2.28	246	2.20	269	2.02	333	1.87	410	....	....	....	....	....	....	....	....	....	....
187	550	2.47	270	2.39	296	2.20	366	2.08	450	....	....	....	....	....	....	....	....	....	....
183	600	2.67	296	2.57	323	2.37	400	2.21	490	....	....	....	....	....	....	....	....	....	....
178	650	2.87	320	2.79	350	2.59	433	2.40	531	....	....	....	....	....	....	....	....	....	....
156	700	3.44	345	3.33	376	3.08	466	2.88	573	....	....	....	....	....	....	....	....	....	....
151	750	3.70	370	3.58	404	3.32	500	....	....	....	....	....	....	....	....	....	....	....	....

NOTE.—To find the size of compressor (piston displacement required) divide the quantity of air obtained by table by the volumetric efficiency of the compressor to be used.

The above table is figured from formula:

$$Va = \frac{h}{0.8\ C \log \frac{H + 34}{34}}$$

Va = Free air per minute actual.
h = Total vertical lift in feet.
H = Submergence in feet.
C = Constant as given in above table.

Starting Pressure = depth of foot piece in well less the static head × 0.434.
Working Pressure = depth of foot piece in well less the pumping head × 0.434 + friction in air + 2 lb. back pressure in nozzle or foot piece.

This table furnished by Ingersoll-Rand Company.

TABLE VI. STEAM AND AMMONIA CONDENSING

1. Cooling tower:		
(a) Water taken from		
(b) Water discharged to		
2. Ammonia condensers:		
(a) Water taken from		
(b) Water discharged to		
3. Steam condensers:		
(a) Water taken from		
(b) Water discharged to		
4. Average temperature water:		
(a) On cooling tower		
(b) Off cooling tower		
(c) Difference		
(d) On ammonia condensers		60
(e) Off ammonia condensers		60
(f) Difference		
(g) On steam condensers		55
(h) Off steam condensers		72
(i) Difference		17
5. Height above pumps:		
(a) Inlet to cooling tower		
(b) Inlet to ammonia condensers		
(c) Inlet to steam condensers		

6. Steam condensed:	B.Hp.-Hr. Condensed	% of Total	
(a) Refrigeration	224.129	4.924	
(b) Electric power and light	231,032	50.76	
(c) Compressed air			
(d) Pumping			
(e)			
(f)			
(g) Total	455,161	100.00	
7. Total M. lb. steam condensed			13,853
8. Gallons condensing water per lb. condensed			6.73
9. Total M. gal. water to steam condenser			93,231

Water Going Over Cooling Tower, Ammonia Condensers and Steam Condensers

10. (a) Avg. head in lb. at pump		
(b) Total M. gal		
11. Charge per M. gal. for water		
12. Total charge for water		
13. Hydraulic hp.-hr		
14. (a) Boiler hp.-hr. for pumping		
(b) Exhaust steam discharged to		
(c) B.hp.-hr. consumed		
15. Kw.-hr. for pumping		
16. Total cost steam for pumping		
17. Total cost electric power for pumping		
18. Total cost labor, supplies, repairs, rent and depreciation for pumping		
19. Total costs for water and pumping		

	% of Charge	Charge for Pumping	Charge for Water
20. Ammonia condensers			
21. Steam condensers			

Water Over Ammonia and Steam Condensers Separately

	Ammonia Condensers	Steam Condensers
22. (a) Avg. head in lb. at pump	46	18.25
(b) Total M. gal. water to condenser	10,064	93,231
23. Charge per M. gal.		
24. Total charge for water		
25. Hydraulic hp.-hr.		
26. Boiler hp.-hr. for pumping		
27. Kw.-hr. for pumping	11,835	44,411
28. Total cost steam for pumping		
29. Total cost electric power for pumping	47.233	1,772.44
30. Total cost labor, supplies, repairs, rent and depreciation for pumping	53.07	507.09
31. Total of costs for water and pumping	525.40	2,279.53

Water Loss by Evaporation

32. Per cent loss over cooling tower		
33. Total M. gal. lost over cooling tower		
34. Value per M. gal.		
35. Total value water lost		
36. Per cent chargeable to ammonia condensing		
37. Per cent chargeable to steam condensing		
38. Charge to ammonia condensing		
39. Charge to steam condensing		

TABLE VI, CONTINUED. STEAM AND AMMONIA CONDENSING

40. Total M. gal. over ammonia condensing		
41. B.t.u. added to cooling water per gal		
42. Total B.t.u. into cooling water		
43. Total B.t.u. from ammonia		
44. B.t.u. taken up by ammonia		
45. M. gal. water evaporated		
46. Value per M. gal.		
47. Total value water lost		

Vacuum Pumps for Steam Condensers

48. Total b.hp.-hr. consumed	8,208
49. Cost per b.hp-hr	0.018197
50. Charge for steam for vacuum pumps	149.36
51. Cost of labor, supplies, repairs, rent and depreciation	
52.	
53. Total cost to operate vacuum pumps	

Steam Condenser Costs

54. Labor to operate and keep condenser clean	$
55. Supplies for operation of condenser	$
56. Repairs to condensers	$
(a)	$
(b)	$
57. (a) Total value of condensers with all connections	$ 14,390.00
(b) Rental charge per annum, 6%	863.40
(c) Depreciation charge per annum, 5%	$ 1,439.00
(d) Rental charge for period	$ 132.83
(e) Depreciation charge for period	$ 221.39
(f) Insurance	$
(g) Taxes	$
58. Total of all steam condensing costs	$ 2,783.11

Subdivision of Steam Condenser Charge

Refrigeration	%	$ 1,370.40
Electric power and light	%	$ 1,412.71
Compressed air	%	$
Pumping	%	$

TABLE VII. GALLONS WATER TO CONDENSE 1 LB. STEAM (THEORETICAL REQUIREMENTS)

Deg. Dif. between Water on and Water off. Condsr.	Temperature of Water off Condenser 90	100	110	120	130	140
12	10.70	10.60	10.50	10.40	10.30	10.20
13	9.87	9.78	9.68	9.5?	.50	9.41
14	9.17	9.08	8.99	8.91	8.82	8.73
15	8.55	8.47	8.39	8.31	8.23	8.15
16	8.02	7.95	7.87	7.79	7.72	7.65
17	7.55	7.48	7.41	7.34	7.27	7.19
18	7.13	7.05	6.99	6.93	6.86	6.79
19	6.75	6.69	6.63	6.56	6.50	6.44
20	6.41	6.35	6.29	6.23	6.17	6.11
21	6.11	6.05	5.99	5.94	5.88	5.83
22	5.83	5.78	5.72	5.67	5.61	5.56
23	5.58	5.53	5.47	5.42	5.37	5.32
24	5.35	5.30	5.25	5.19	5.15	5.09
25	5.13	5.09	5.03	4.99	4.94	4.89
26	4.94	4.89	4.84	4.80	4.75	4.70
27	4.75	4.71	4.66	4.62	4.57	4.53
28	4.58	4.54	4.50	4.45	4.41	4.37
29	4.43	4.38	4.34	4.30	4.26	4.22
30	4.28	4.24	4.20	4.15	4.12	4.08
31	4.14	4.10	4.06	4.02	3.98	3.95
32	4.01	3.97	3.93	3.90	3.86	3.82
33	3.89	3.85	3.82	3.78	3.74	3.71
34	3.77	3.74	3.70	3.67	3.63	3.60
35	3.67	3.63	3.60	3.56	3.53	3.50
36	3.57	3.53	3.50	3.46	3.43	3.40
37	3.47	3.44	3.40	3.37	3.34	3.31
38	3.38	3.35	3.31	3.28	3.25	3.22
39	3.29	3.26	3.23	3.20	3.17	3.14
40	3.21	3.18	3.15	3.12	3.09	3.06
41	3.13	3.10	3.07	3.04	3.01	2.98
42	3.06	3.03	3.00	2.97	2.94	2.91
43	2.99	2.96	2.93	2.90	2.87	2.84
44	2.92	2.89	2.86	2.83	2.81	2.78
45	2.85	2.83	2.80	2.77	2.75	2.72
46	2.79	2.76	2.74	2.71	2.69	2.66
47	2.73	2.71	2.68	2.65	2.63	2.60
48	2.67	2.65	2.62	2.60	2.57	2.55
49	2.62	2.60	2.57	2.55	2.52	2.50
50	2.57	2.54	2.52	2.49	2.47	2.45
51	2.52	2.49	2.47	2.45	2.42	2.40

TABLE VIII. GALLONS OF CONDENSING WATER PER TON OF REFRIGERATION

Cond. Press., Lb. per Sq. In. Gage	Corresponding Temp. Deg. F.	60 Deg. F. Water — Water per Ton of Refg. Gal. per Min. Range Deg. F.	Suct. 15	Press. Lb. 20	Gage 25	70 Deg. F. Water — Water per Ton of Refg. Gal. per Min. Range Deg. F.	Suct. 15	Press. Lb. 20	Gage 25	80 Deg. F. Water — Water per Ton of Refg. Gal per Min. Range Deg. F.	Suct. 15	Press. Lb. 20	Gage 25	90 Deg. F. Water — Water per Ton of Refg. Gal. per Min. Range Deg. F.	Suct. 15	Press. Lb. 20	Gage 25
126.4	75	10	2.90	2.85	2.80	..	....	....	....	..	....	....	....	..	....	....	....
131.4	77	12	2.40	2.35	2.30	..	....	....	....	..	....	....	....	..	....	....	....
136.6	79	14	2.05	2.00	1.95	..	....	....	....	..	....	....	....	..	....	....	....
141.8	81	16	1.85	1.80	1.75	6	4.87	4.80	4.70	..	....	....	....	..	....	....	....
147.2	83	18	1.65	1.60	1.55	8	3.67	3.60	3.55	..	....	....	....	..	....	....	....
152.7	85	20	1.47	1.45	1.41	10	2.95	2.90	2.85	..	....	....	....	..	....	....	....
158.3	87	22	1.35	1.32	1.30	12	2.45	2.40	2.36	..	....	....	....	..	....	....	....
164.1	89	24	1.25	1.22	1.20	14	2.10	2.05	2.00	..	....	....	....	..	....	....	....
170.1	91	26	1.15	1.13	1.10	16	1.90	1.85	1.80	6	5.00	4.90	4.80	..	....	....	....
176.2	93	28	1.08	1.05	1.03	18	1.68	1.63	1.60	8	3.75	3.70	3.65	..	....	....	....
182.6	95	30	1.00	0.99	0.97	20	1.50	1.48	1.45	10	3.00	2.95	2.90	..	....	....	....
189.1	97	32	0.95	0.93	0.90	22	1.38	1.35	1.32	12	2.52	2.48	2.43	..	....	....	....
195.7	99	..	....	....	....	24	1.28	1.25	1.22	14	2.13	2.10	2.06	..	....	....	....
202.5	101	..	....	....	....	26	1.17	1.15	1.13	16	1.92	1.85	1.83	6	5.10	5.00	4.90
209.5	103	..	....	....	....	28	1.10	1.07	1.05	18	1.70	1.67	1.63	8	3.83	3.75	3.70
216.5	105	..	....	....	....	30	1.02	1.00	0.98	20	1.53	1.50	1.46	10	3.07	3.00	2.95
223.7	107	..	....	....	....	32	0.95	0.93	0.92	22	1.40	1.37	1.35	12	2.57	2.51	2.47
231.1	109	..	....	....	....	..	....	....	....	24	1.30	1.28	1.25	14	2.19	2.13	2.10
238.7	111	..	....	....	....	..	....	....	....	26	1.20	1.16	1.15	16	1.95	1.90	1.88
246.5	118	..	....	....	....	..	....	....	....	28	1.12	1.08	1.06	18	1.74	1.69	1.65
254.5	115	..	....	....	....	..	....	....	....	30	1.05	1.02	1.00	20	1.56	1.53	1.50
262.7	117	..	....	....	....	..	....	....	....	32	1.00	0.98	0.95	22	1.43	1.40	1.37

TABLE IX. CALCULATON OF NET CHARGES TO PLANT

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	Steam		Refrigeration		Power and Light — Power		Power and Light — Light		Compressed Air		Water — Purchased		Water — Local	
	B.Hp.-Hr.	Cost	Tons	Cost	Kw.-Hr.	Cost	Kw.-Hr.	Cost	M.Cu.Ft.	Cost	M. Gal.	Cost	M. Gal.	Cost
Total net produced.....	1,749,594	$31,837.30	9,926	.....	258,189	.....	.....	.....	5,799	.....	14,694	.....	96,120	.....
No. units used for following:														
Steam.....	.....	.....	.....	.....	9,023	.....	.....	.....	5,799	.....	5,936	.....	.....	.....
Refrigeration.....	392,319	.....	.....	.....	1,512	.....	.....	.....	.....	.....	.....	.....	.....	.....
Power and light.....	231,032	.....	.....	.....	2,177	.....	.....	.....	.....	.....	.....	.....	.....	.....
Compressed air.....	8,542	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Purchased water.....	4,688	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Local water.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
Ammonia condensers.....	.....	.....	.....	.....	11,835	.....	.....	.....	.....	.....	.....	.....	.....	.....
Steam condensers.....	8,208	.....	.....	.....	44,411	.....	.....	.....	.....	.....	.....	.....	96,120	.....
Total for motive power.	644,789	11,733.22	.....	.....	68,958	.....	.....	.....	11	.....	5,936	.....	96,120	.....
Total to plant.....	1,104,805	20,104.08	9,926	.....	189,231	.....	.....	.....	5,788	.....	8,758	.....	.....	.....

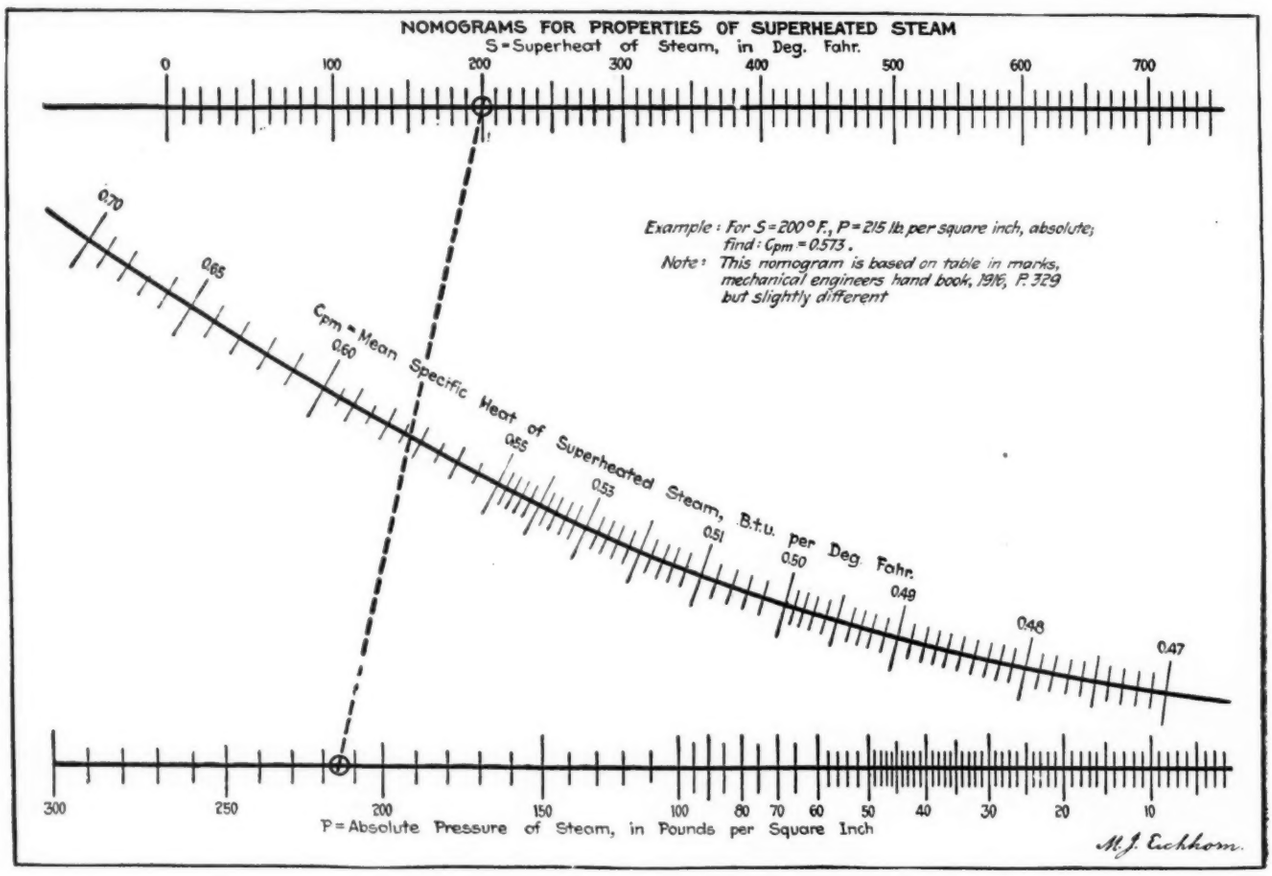
Specific Heat of Superheated Steam

By M. T. EICHHORN

A NOMOGRAM that is of assistance in extending the present steam tables is shown on this page. It gives the mean specific heat of superheated steam when the absolute pressure and degree of superheat are known. Having the mean specific heat and the properties of saturated steam at a certain pressure, the corresponding properties of superheated steam are obtained by means of the approximation formulas given in Marks' "Mechanical Engineers Handbook" and elsewhere. This nomogram is constructed by approximation methods from a numerical table in the book mentioned (1916 ed., p. 329). Readings are obtained from this nomogram by taking a point on each of the outside scales, corresponding to the given data, and connecting them with a straight line, preferably scratched on a piece of transparent celluloid, which line then intersects the curve in the middle at a point representing the specific heat.

The principal superiority of such a nomogram over a numerical table consists in the ease of finding values intermediate between those tabulated—which is a very laborious process in a numerical table of double entry—and also in the fact that when the nomogram has been drawn, any one of the three variables may be considered as the unknown quantity. The results given by the nomogram are slightly different from those of the table that is given in Marks', but in view of the remaining discrepancies of the basic data in the region under consideration, this deviation may be regarded as of no practical importance.

[A nomogram showing the temperature and pressures of steam up to the critical point, in both English and metric units, was published in *Power* May 17. Nomograms dealing with other properties of steam will appear in succeeding issues.—EDITOR.]



Power Transmission by Wire Rope

Application and Arrangement of System, Type and Size of Sheaves—Rope Construction, Driving Tension, Velocities and Horsepower Transmitted—Idle Sheaves and Takeup Devices

By A. J. DIXON

SYSTEMS of power transmission by wire rope are of limited application. Practically the only situations in which such systems are installed are where comparatively small amounts of power are to be conveyed, out of doors, between separate buildings of a manufacturing plant, or where power must be distributed over great distances in different directions from a common source, as in the manufacture of explosives. In this industry it is desirable that the various processes be carried on in separate buildings, spread out over a large area so as to isolate the buildings from one another. It is also sought to locate the source of power, if a steam or internal-combustion engine is used, as remote as practicable from where the power is to be utilized. A system of transmission by wire rope answers admirably for this purpose. Where the distance traversed does not exceed about 400 ft. the system consists simply of an endless wire rope looped around the grooved peripheries of a driving and a driven sheave. It is merely the substitution of wire rope and grooved sheaves for leather, canvas or rubber

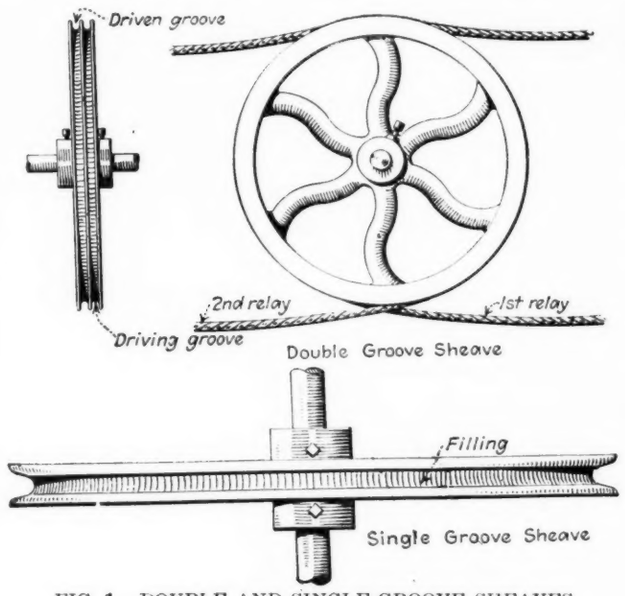


FIG. 1. DOUBLE AND SINGLE-GROOVE SHEAVES

belting and flat-faced pulleys. The power is transmitted from the driving sheave to the driven sheave by virtue of the traction that is developed by the rope as it runs on the grooved peripheries of the sheaves.

But if power must be conveyed to a greater distance than 400 ft. the system will generally consist of two or more wire ropes looped successively around driving and driven sheaves. The power is then conveyed through a series of relays from the source to the point of utilization. A driving sheave may be mounted on the same shaft with the driven sheave of the first relay, and so on with the second and third and succeeding relays. The sheaves that are intermediate between the first and last sheaves in the series may have two grooves, as shown in Fig. 1. Power is then received from the rope running in one of the grooves and is given off to the rope running in the other groove.

The sheaves should be of as large diameter as practicable so as to insure a high peripheral speed and therefore a high rope speed relative to the speed of the shaft. The peripheral speed should not, however, exceed about

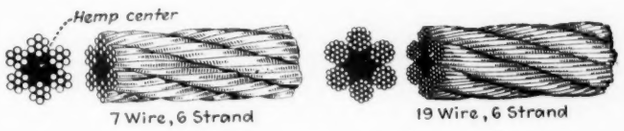


FIG. 2. WIRE-ROPE CONSTRUCTION

5,000 ft. per min., since the limit of safety for sheaves of cast-iron construction is approached at this speed. The higher the velocity of a wire rope the smaller the size of rope required to transmit a given amount of power. Also, the greater the diameter of the sheaves the less the bending stresses in the wires of the rope. Excessive bending stresses may be obviated by using sheaves having diameters at least 100 times the diameter of the rope if a rope of 7-wire 6-strand construction is used, or at least 60 times the diameter of the rope with 19-wire 6-strand construction. Both classes of rope are shown in Fig. 2. Ample sheave diameter is also necessary to prevent the upper stretch of the rope from sagging far enough to run foul at the lower stretch.

Sheaves Should Be Accurately Balanced

On account of the high speed required of the sheaves special care should be exercised to see that they are accurately balanced. It is also essential that the sheaves be in perfect alignment and that the shafts upon which they are mounted are exactly parallel. It will be impossible for the rope to run smoothly if these conditions are not observed. When the alignment of the sheaves is defective the rope will inevitably jump, jerk and sway, to the detriment of both itself and the material with which the grooves of the sheaves are filled.

It is also necessary that the sheaves have sufficient mass to produce a flywheel effect during fluctuations of the load. It requires but little impulse to start a wire driving rope jumping and swaying. Such impulse might be due to a momentary checking of the speed caused by a sudden increase of load. If the sheaves are massive enough their momentum will neutralize these effects of variations in the loads. Swaying of a running wire rope is both dangerous and destructive. The rope will wear against the iron surfaces of the grooves and may also run foul of surrounding objects. Sometimes it may be necessary to smooth out the irregular tendency of a fluctuating load by mounting a flywheel on the driven shaft alongside the sheave.

Sheaves made wholly of cast iron are generally used when the requisite diameter is 12 ft. or less. For larger diameters the sheaves are built up commonly with cast-iron rims and wrought-iron arms. When the diameter

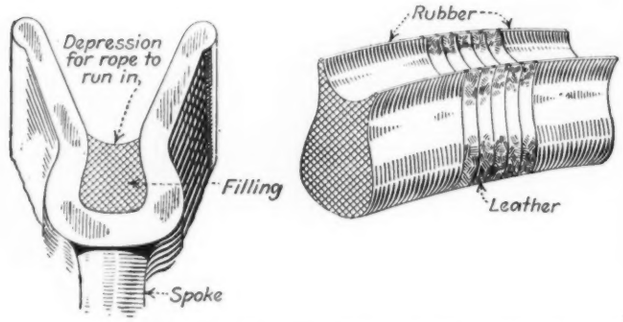


FIG. 3. GROOVE OF SHEAVE WITH RUBBER AND LEATHER FILLING

is less than 10 ft. the sheaves usually are cast in one piece. Above that size they are made in halves, which are bolted and clamped together at the hub and rim.

Grooves of wire-rope sheaves are of the form shown in Fig. 3. The V-shaped groove merges with a flaring chamber at the bottom, which is filled with a material

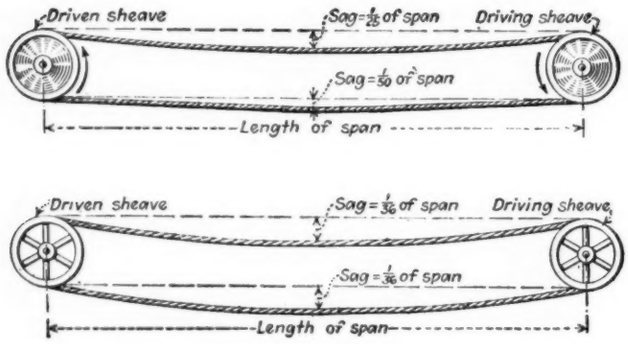


FIG. 4. SAG OF RUNNING AND STANDING ROPES

that insures good adhesive friction. The rope runs on this filling. Under normal conditions it never touches the iron surfaces of the groove. Blocks of rubber, alternating with pieces of sole leather, are commonly used for the fillings. Each segment of rubber or leather is dipped into molten pine tar before it is driven into the groove. Blocks of hardwood also may be used. The purpose of the filling is, primarily, to secure good traction for the rope. The wires of the rope bite into the slightly yielding surface of the filling and thus obtain a firm pulling grip as the rope runs around the sheave. But in addition to this the filling also tends to diminish the wear on the rope.

The middle of the filling should be slightly hollowed out or depressed, as shown in Fig. 3. This will tend to keep the rope riding in the middle of the groove instead of rolling from side to side and wearing against the iron surfaces. To initiate the formation of a proper depression for the rope to run in, a shallow V-shaped groove should be turned in the face of the filling when it is newly inserted.

To transmit a given amount of power the driving tension is applied mainly by the weight of the rope. The same size of rope is required to give the necessary weight, whether the material be steel of high tensile strength or the weaker wrought iron. Hence in most cases an iron rope will suffice as well as one of steel, since the size will always insure ample strength. The only advantage in using a steel rope is the greater durability obtained. A steel rope having the same strength as a wrought-iron rope of a given size would be smaller than the iron rope, and if used would have to be stretched tighter to give the requisite driving tension. This would be detrimental to the filling in the sheave groove, as the tightly drawn rope would tend to lacerate and cut it out.

Wire ropes used in transmitting power are generally of the construction shown in the left view of Fig. 2, consisting of six strands, each containing seven wires, twisted around a hemp central core. When it is necessary to use sheaves of comparatively small diameter, however, ropes containing nineteen wires to a strand may be preferable. The coarser construction is advantageous inasmuch as the seven-wire strand will withstand the greater wear. This is due to the fact that the wires are over 60 per cent larger. Such a rope requires larger sheaves to run on, but usually this is not a disadvantage, since employment of a proper peripheral speed will insure an ample sheave diameter in ordinary cases.

The two principal factors determining the amount of power transmitted by a wire rope are the tensile stress

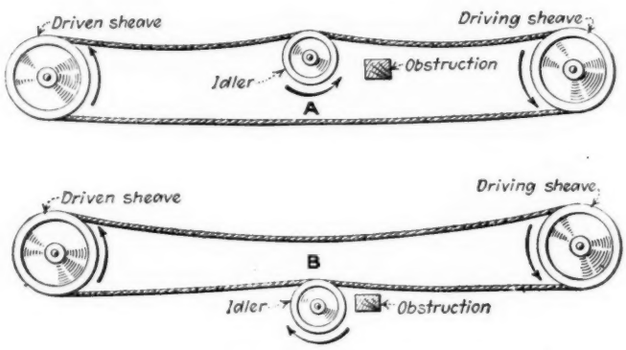


FIG. 5. THE PLACING OF IDLERS

in the rope and the velocity. Since the velocity is generally high, the direct pull on the rope when transmitting a given amount of power is comparatively low. Hence the main consideration in selecting a rope is that it will have enough weight to keep it tight against the sheaves.

In a wire rope the driving tension for power transmission is governed mainly by the weight of the rope,

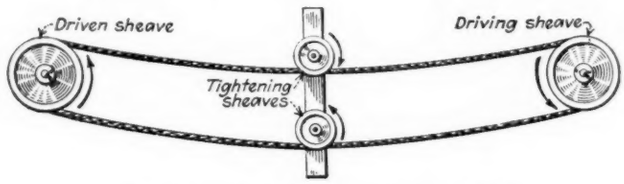


FIG. 6. SHEAVES TO TIGHTEN ROPE

the distance between the sheaves and the sag or deflection of the rope. When the rope is not running, a deflection of both the upper and lower stretches of the drive equal to about $\frac{1}{36}$ of the distance or span between the sheave centers has been found to suffice under ordinary conditions. For example, if the sheaves are 250 ft. from center to center, the standing deflection should be approximately $250 \div 36 = 7$ ft.

Assuming that the standing deflection of the rope is $\frac{1}{3}$ of the span, then if the driving sheave revolves in a direction that will put the pull on the lower stretch of the rope, as it properly should do, the lower side of the drive will tighten up, and when a proper maximum speed is attained the deflection on this side will have dwindled to about $\frac{1}{50}$ of the span. At the same time the upper stretch of the rope will gradually loosen and sag until its deflection has increased to about $\frac{1}{25}$ of the span. The sags of the rope when standing and when running are shown in Fig. 4.

There is a great advantage in exerting the pull of the driving sheave on the lower stretch of the rope. When the pull is on this side the upper stretch sags and gives a much greater arc of contact with the sheaves and, consequently, greater traction than when the pull is on the upper side. In the latter case the rope would sag mainly on the lower side and away from the sheaves, thus giving a smaller arc of contact.

While the changes in the deflections of the two sides of the drive are occurring, as the rope speeds up the tension in each side is likewise changing. The tension diminishes in the upper stretch of rope and correspondingly increases in the lower stretch. The effective tension in the rope, or the pull exerted by the driving sheave on the rope, and of the rope on the driven sheave, will then be the difference between the tension of the two sides. This net tension, for the proportion of standing deflection previously suggested, is approximately equal to three times the weight of the rope suspended between the sheaves on either side of the drive. Hence the amount of power transmitted by a wire rope drive may be computed by the formula,

$$P = \frac{WLV}{11{,}000}$$

wherein P is the horsepower transmitted, W is the weight of the rope in pounds per foot, L the span of the drive, or the distance in feet between the sheave centers, and V is the velocity of the rope in feet per minute.

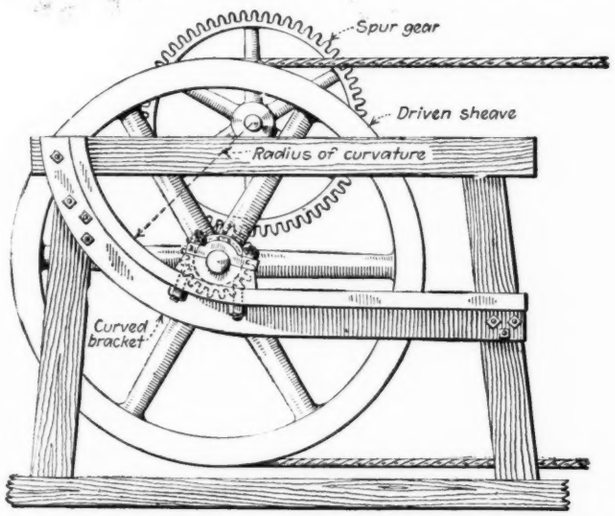


FIG. 7. TAKING UP SLACK BY ADVANCING BEARINGS ON CURVED BRACKETS

Suppose, for example, 7-wire, 6-strand, $\frac{1}{2}$-in. wire rope, running on 6.5-ft. sheaves, is transmitting power from a lineshaft in one building in a group constituting a manufacturing plant to a parallel lineshaft in another building in the same group. The speed of the driving shaft is 160 r.p.m., and the distance between the shafts is 140 ft. What amount of power is transmitted?

According to the manufacturer's tables a 7-wire, 6-strand, $\frac{1}{2}$-in. wire rope weighs 0.39 lb. per lineal foot. The velocity of the rope is $160 \times 3.1416 \times 6.5 = 3{,}267.3$ ft. per min. Hence, by the formula, the amount of power transmitted is

$$P = \frac{WLV}{11{,}000} = \frac{0.39 \times 140 \times 3{,}267.3}{11{,}000} = 16.22 \text{ hp.}$$

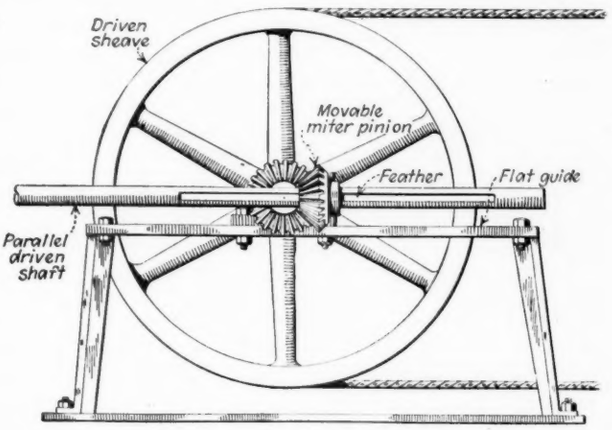


FIG. 8. ANOTHER METHOD OF TAKING UP SLACK

Suppose it is desired to transmit 65 hp. a distance of 300 ft. by wire rope. The sheaves are to be 7 ft. in diameter, and the driving sheave is to run at 155 r.p.m. What size of 7-wire, 6-strand rope is required?

The velocity of the rope will be $7 \times 3.1416 \times 155 = 3{,}408.6$ ft. per min. Hence by transposition of the formula the weight of the rope equals

$$W = \frac{11{,}000\,P}{LV} = \frac{11{,}000 \times 65}{300 \times 3{,}408.6} = 0.7 \text{ lb. per ft.}$$

The nearest value to this in the manufacturer's tables of 7-wire, 6-strand rope is 0.75, which is the weight per foot of a $\frac{13}{16}$-in. rope. Hence a $\frac{13}{16}$-in. rope is required.

There is a continual loss of energy in a wire-rope drive on account of the friction between the wires as the rope bends around the sheaves. Added to this is the counteracting effect, on the adhesion between the rope and the peripheries of the sheaves, of the centrifugal force developed by the running rope. But the aggregate of these losses is generally negligible under the usual conditions of operation.

It is sometimes necessary to guard against contact between the rope and obstructions that intervene between the driving and driven sheaves. Such protection is secured by installing idle sheaves, as shown in Fig. 5. The use of an idler on the slack side of the drive is objectionable, inasmuch as it tends to reduce the effective tension of the rope. When an idler is placed in the middle of the span, as shown at A, it reduces the net driving tension and the drive is then capable of transmitting less than it could otherwise. The deficiency may be compensated, however, by increasing the size of the rope, thus securing greater weight. When an idler is installed on the pulling side of a drive, as shown at B, the amount of power transmitted will not be altered.

When it is necessary to install idler sheaves their diameters should be as large as practicable—at least

one-half that of the driving and driven sheaves. They should be in alignment with the main sheaves, for if they are not the rope will be continually subjected to spasmodic jerking strains which, in conjunction with the abrasive wear, will result in early disruption of the rope by breakage of the wires.

Sheaves for increasing the tension in a rope drive by tightening the two sides of the drive, as in Fig. 6, are sometimes installed. Their use is, however, generally objectionable. If a tightener is necessary in transmitting the requisite amount of power, the fact usually indicates that the rope is overloaded and overstrained.

A new rope will stretch considerably for some time after it is installed, but after the initial stretch has been cut out of the rope and it has been respliced it should, under favorable conditions, run for many months before shortening is again necessary. But if the stretch continues at such a rate as to call for the use of tightening sheaves, substitution of a larger and heavier rope will in most cases be preferable.

Taking Up the Slack

Devices for taking up the slack of a wire transmission rope are sometimes installed. These are designed to obviate the shortening of the rope by cutting and resplicing. With the device shown in Fig. 7, the driven shaft rests in bearings that are mounted on curved brackets. The arc of curvature of the brackets is parallel to the pitch circle of the spur wheel that engages with the pinion on the end of the shaft. The slack is taken up by loosening the stud bolts that clamp the bearing boxes to the curved brackets, pushing the boxes outward to a new location, and then retightening the bolts.

With the device shown in Fig. 8 the bearing boxes of the driven shaft are mounted on flat guides. The power is transmitted through miter gearing to a shaft that extends in a parallel direction with the rope. The miter pinion on this shaft is locked to the shaft with a long feather and two setscrews. The slack is taken up by loosening the setscrews that hold the driven pinion and the bolts that hold the bearing boxes, pushing the boxes and pinion outward to new locations and then retightening the screws and bolts.

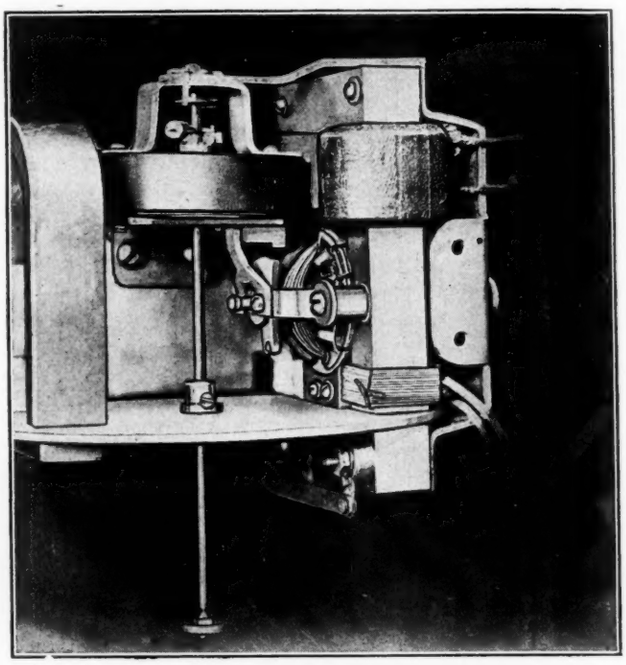
Graphic Kilovolt-Ampere Demand Meter

It is generally recognized that the cost of serving a power customer is affected by the power factor of the load, and rates are now being put into effect which take the power factor into account. The insistent demand for a kva. meter has arisen from the fact that it is mainly the "investment" or "readiness to serve" cost that is affected by power conditions, so that the demand charge should be based on the customer's maximum demand measured in kilovolt-amperes. The kva. meter makes this possible.

On account of the great demand for a kva. measuring instrument, practically all electrical-instrument manufacturers have been working on this problem, and the Esterline-Angus Co., of Indianapolis, Ind., announces that it has succeeded in producing a simple, practicable kva. instrument, which is now being placed on the market.

This meter is of the induction type and can be made in the graphic, integrating, indicating or interval-demand type. The graphic has been developed first. The measuring element is shown in the figure and is about the size of the element of a polyphase wattmeter. Each single-phase element consists of a pair of laminated poles, carrying exciting windings, between which is pivoted a small armature, which maintain a fixed phase relation between the field produced by the current windings and that produced by the voltage windings.

The instrument is equally accurate throughout the entire range of power factor from unity to zero, either leading or lagging, and records the true kva. of the circuit, regardless of the degree of unbalance. The three-phase instrument indicates the sum of the products of current per phase times corresponding voltage to neutral, which is the true kva. of a three-phase circuit. The accuracy of the instrument is well within the established limits for electric meters, regardless of commercial variations in voltage, frequency and wave form. Meters of this type have been designated and completely developed in the portable, switchboard and wall types.



MEASURING ELEMENT OF KVA. METER

According to the N. E. L. A. Meter Committee report, information obtained in previous years appeared to show no definite change in the characteristics of instrument transformers on periodic tests after being in service a number of years. Re-tests made during the last year further confirmed this opinion. With modern transformers made of high-grade silicon steel, there is less possibility of any change taking place in service, that would seriously affect the transformer characteristics and not be apparent to external examination than with the earlier types, so that the testing period of transformers so constructed can be safely extended. One possibility of change in current transformers may be due to magnetization. This, however, can be corrected by demagnetizing the transformer, if it is known to have been open-circuited or heavily overloaded.

EDITORIALS

More Facts Concerning Pulverized Coal

RARELY does one find compressed into so small a space so much food for thought and valuable information as given by Messrs. Kreisinger and Blizard in their contribution on boiler tests with pulverized coal, prepared for the current meeting of A. S. M. E. Much-needed information as to the use of pulverized coal is afforded through their tests, which are of an accuracy and interest far above the average.

This work is perhaps the first series of tests that has ever been made in which the powdered fuel was actually weighed. In this respect alone it is notable. To have the unaccounted for energy never greater than about three per cent of the total is most satisfying to the engineer who wishes really to understand just what is going on in the steam system that he is investigating.

One of the most striking points brought out was that relating to the fineness of pulverizing. If one may look forward to less drastic effort in gaining fine material for "pulverized" coal, much wider application and much greater economy may be anticipated. It is particularly gratifying to know that the completeness of combustion is a matter of proper furnace design rather than of the fineness after a reasonable degree of pulverization has been reached.

That moisture is not a constituent of coal which must be altogether (or practically altogether) prohibited when pulverized fuel is used is also encouraging. Of course the coal must be dried sufficiently to prevent packing in storage or transportation, else mechanical troubles entirely foreign to the furnace will be met. But if that considerable part of the expense of the fuel preparation which goes for drying the fuel to one per cent moisture can be saved, the inconvenience and perhaps even the danger of handling the fine coal may be reduced and costs of preparation, of course, will be materially lessened. When fine coal is introduced into the combustion space of a furnace accompanied by a small percentage of moisture, say three to six per cent, the carbon, as soon as it reaches incandescence, reacts quickly with the steam formed by the moisture to produce hydrogen and carbon monoxide. In other words, a water-gas reaction apparently takes place. The problem, then, is simply to complete the combustion of these two very easily burned gases, and this is simplified by the fact that it is gas and not a solid that is being burned. It is probable that in the presence of reasonable percentages of moisture such as those suggested, the initial stages of combustion take place within a shorter period and smaller firebox spaces will serve than those required if all the oxygen for preliminary combustion must come from air diffusing into contact with each separate coal particle. It is not surprising in view of these fundamental facts, which are all too often ignored, that the results reported by the authors of this recent report show there is no tendency for greater loss due to incompleteness of combustion in the presence of small percentages of water in the fuel.

As a matter of fact it is quite likely that we have been worrying about a relatively small weight of water in the coal and all the time forgetting that the air entering the furnace carried tremendous quantities of water because it is by no means so dry as most of us would think. Each pound of solid fuel that we burn requires many times its weight of air to complete the combustion, and the air almost invariably contains one to one and one-half per cent by volume of water vapor.

It is to be hoped that this type of investigation can be carried further by these investigators or others so that more may be learned as to the capacity of boiler installations with pulverized fuel.

Tendencies in Electrical Controller Practice

PROBABLY in the development of no other class of equipment has there been so great a variety of different types produced as in controllers for electric motors. This is as might be expected on account of the wide variation in the application of the electric motor. Although there are a large number of drives to which a standard class of controllers is fairly well suited, there are a multitude that require special control equipment designed for the service. In fact, the general statement may be made that "before an electric drive can be successful, there must be not only a motor suited to the application, but also a controller that will function properly under the conditions imposed by the load." These conditions may vary from starting and stopping a motor once or twice a day under light load to several hundred times a day under all manner of load conditions, as for example, heavy passenger or freight elevator service. The opening and closing of the circuit under all these conditions must be done with a contactor, and it is upon the proper design of this contactor that the successful functioning of the controller in no small degree depends.

On the contacts through which the circuit is made and broken there is inherently more or less burning. These contacts have been developed in three general forms—butt, sliding and rolling. All three have been tried with varying degrees of success. For a considerable period the tendency was to use a combination copper and carbon butt contact, but with the development of the rolling contact the trend is toward copper-to-copper, which has a much higher degree of conductivity and consequently can be made much lighter in weight for a given current capacity. Since this type has a rolling action not only during the closing, but also when opening the circuit, there is little liability of its fusing and holding the circuit closed. With the rubbing contact so commonly used in cylinder-type controllers, lubrication is almost imperative in almost all applications if cutting of the contacts is to be prevented. To get away from this, some manufacturers are designing

these controllers so that rolling contacts can be applied. At one time it was thought by many that a copper-to-copper contact for opening and closing a circuit was impractical, but the development of the rolling contact in conjunction with a properly designed blowout coil has practically eliminated all the former difficulties.

From the operator's viewpoint no tendency is of more interest than that toward standardization. In some of the controllers built a few years ago, it was not uncommon to find two controllers of the same type and capacity the parts of which were not interchangeable. Manufacturers now build standard contactors, master switches, etc., which are incorporated into the various controllers. These standard units meet the specifications of the American Institute of Electrical Engineers and the Electric Power Club, so that the performance of the various units furnished by different manufacturers is on the same basis. One of the advantages of using standards is reliability of operation. New designs usually need improvement and correction after operating in service. The wider the range of application the more thorough is the test of such apparatus and the greater the reliability. The users of standard apparatus are not limited to specifications, but can obtain accurate information regarding the performance of this apparatus and can decide between various sizes and types so that the selection will best suit the particular application.

Standard apparatus, in turn, is built up from standard parts, which makes it easier to obtain repairs and permits the stocking of these repair parts where a number of controllers are in use. Standard apparatus can be built more promptly than special and at a considerable saving in first cost. Economical reasons alone should dictate the use of commercial standards wherever practicable. Controllers are much simpler and more durable, when standards are adopted. It permits the accumulation of test and operating data to determine what details may be improved and what complications may be omitted.

Much is being accomplished just now in the study of special applications and the adapting of the electrical apparatus to the requirements of the drive. Examples of this are the electric elevator, the printing press, the paper machine, power-station auxiliaries and a variety of other applications. By making an engineering study of the requirements of the drive, the proper motors and control equipment can be selected and any special features required of the control can be developed.

Brains in the Power Plant

ENGINEERING, including the branch that is concerned with power-plant work, is, more than any other department of industry, well-nigh an exact science. In engineering most effects can be traced back to the underlying causes. Nevertheless, in thousands of plants both large and small no great effort is made to get at the basis of the plant troubles. The engineer, instead, repairs the damage and expectantly awaits another manifestation of plant defect with no attempt to locate the origin of the trouble. For example, all know the type of engineer who will repeatedly repair a leaky pipe joint, renew gaskets, tighten the bolts and cuss the pipe, the plant and the world in general with never a thought to the lack of expansion loops or joints in the long steam line. A hundred similar acts that denote the lack of functioning of brains could be listed. Unfortunately, such "engineering" is not confined to small isolated plants, but exists even in some that are quite show places.

In the old days when coal was cheap, it mattered little how inefficiently a power plant operated. Power costs made up but a small part of the production costs of any article. Profits of manufacturing were ample to cover the extravagances that might exist in the factory. Today this is all changed. In the strife of competition production costs are being cut to the bone. The one or two cents a barrel that might be lost through inefficiency may, to a flour mill, represent the difference between profit and loss. In the endeavor to lower costs the power plant is undergoing investigation, and the engineer who fails to meet expectations is in danger of being shelved.

If an engineer hopes to increase his earning power, he must show that he can make a profit for his employer. This cannot be done by the use of brawn. Brains alone is the means whereby the engineer can hope to progress.

Plant Environment and Better Help

MOST men are creatures of sentiment and have an innate love for the beautiful, for nature and the brighter side of life. This is more highly developed in some, but there are few indeed among even the illiterate that do not love the better things of life. If we surround a man with an environment of pleasing sights that make the place wherein he labors more inviting, his tasks become more pleasing and his labors less irksome. He glories in the very things that are placed before his eyes and loses in great measure that state of mind that places him on a par with a machine.

Take any man and shut him up in a dark, dingy, insanitary shop or plant and then hope for the best that is in him and the result will be disappointing. His nature will rebel against this unnatural environment. It matters not how optimistic he may be, the effect of this condition will become apparent and he will not be capable of his best work. He becomes, in a measure, like a caged beast and loses that natural instinct to go forward, that desire to do his best and that incentive that is otherwise a part of his makeup. Besides, it fosters a disposition to become careless.

There are many plants so situated that their environment can be materially improved, and the lot of the employees made better. Opportunities for doing this are not confined to the interior of the plant but extend to the exterior as well. The importance of the idea is being more fully grasped and the future augurs well, not alone for the employees but for the employers also. The increased power for better work that is thus made possible is seen in greater efficiency of the men.

Do these things pay the employer? Do they make for a better class of employees and are the men not grateful because of the things that have been done for their betterment? What is the answer and what is there to say to the contrary? Men are human, after all, and as such they crave the better things in life, even while they labor. There is so much to be said in favor of a more pleasing environment and so little to be offered in favor of the contrary that the matter offers really no room for argument. Efficiency, interest, and loyalty are materially increased where these have been made possible.

CORRESPONDENCE

Application of Cooling Coils to Coal Storage

Cooling coils are applied to cold-storage rooms, condensers, refrigerators and various other things, so the question naturally presents itself, Why not apply them to the coal-storage pile? Of course it would not be practical to use a refrigerating machine for keeping the coal pile cool, but it would be practical to have a network of coils uniformally distributed throughout the coal pile, through which water could be circulated, thus carrying off the heat generated in the storage pile and so prevent the coal from reaching the ignition point.

In a certain plant several thousand tons of coal was stored and a great deal of trouble was experienced from spontaneous combustion. Through the center of this storage pile, a 2-in. water line ran, carrying water to a screen used for washing ore. It was noted that ignition did not occur at any point within 18 in. of this pipe. Later, an experimental storage was built, one-half of which was equipped with cooling coils and the other half without the coils. Both were filled with coal from the same mine, of about the same size, being what was termed nut and screenings. Each storage compartment had a capacity of about 3,000 tons.

The water-cooled storage room was provided with 2-in. coils which were run vertically instead of horizontally and were placed 18 in. apart. Headers were run across the top and bottom for the inlet and outlet of the water. Water from the mine was pumped through these coils at a temperature of about 60 deg. F. Thermometers were placed in the inlet and outlet so that the difference in temperature, if any, could be noted.

For the first thirty days after filling the storage, no rise in the temperature of the water passing through the coils was noted, but after forty-five days the temperature rise in the water after passing through the coils began to be appreciable, and at the end of sixty days there was a difference of from 5 to 15 deg.; at the end of ninety days from 7 to 19 deg.; after which the temperature difference began to decrease. At the end of 180 days there was very little if any difference in the temperatures of the inlet and outlet water. Ignition did not occur at any point in this storage pile.

The storage that was not equipped with the cooling coils gave considerable trouble, having to be moved several times during the 180-day test.

It may not be practical to use cooling coils in all cases where coal must be stored, and in any case there are certain objections to them, but there are also objections to any method now being used. The main objection to the coils is the difficulty of reclaiming the coal when coils are installed, but if they are run vertically and about 18 in. apart, and if it is possible to have a tunnel underneath the storage with chutes leading to the tunnel, it will be found that the coils give very little trouble in the reclaiming of the coal.

Logansport, Ind. WILLIAM M. MCNEILL.

Proposed Home-Made Engine Stop

In the April 12 issue of *Power*, page 596, J. Redmond showed a sketch of a proposed home-made engine stop and gave some explanation of it. The overspeed part of this device is somewhat new to me, but the chief objection to it is that as the wear takes place the engine will shut down at a higher rate of speed, and in addition it would be difficult to figure the jump of the roller.

The main objection to this engine stop, however, lies in the fact that any device winding the throttle shut is not a safety device. At this point enters the time element. There have been a number of engine disasters with this type of stop, where, after the explosion the throttle was found to be shut, showing that the device worked but not quickly enough.

The only safe engine stop, in my opinion, is one with a substantial, well-designed, quick-closing valve in the steam line close to the throttle—a valve free of internal dashpots and sliding stems. This type of valve closes off every particle of steam before the flywheel has made a half-revolution at a dangerous rate of speed.

The simplest type of overspeed device possessing, I believe, the least chance of failure, is one having a speed band attached to the mainshaft. This band carries a pin attached to a curved lever arm with a counteracting spring. The faster the shaft revolves the farther out this pin moves. A stand carrying a tripping attachment having a vernier adjustment, is mounted in such a position that when the speed increases from 2 to 3 per cent beyond the point for which the stop is set, the pin on the speed band strikes the trip, which instantly cuts off all steam to the engine.

With a stop of this kind, if a number of remote-control stations are required, electric attachments can be added to the tripping attachment and a solenoid attached to the quick-closing valve. I am a firm believer in using a closed circuit, for then the entire system is always in test and any corroded wires will shut down the engine and the trouble can be located and remedied. H. E. PIRATT.

Montvale, N. J.

What Caused This Pipe Corrosion?

In the issue of *Power*, April 12, page 596, Mr. Forbes asks, "What Caused This Pipe Corrosion?" The statement says, the pipes are horizontal and overhead. It follows that there would be an accumulation of dust on the top of all the piping.

This corrosion is, I believe, a case of obscure electrical action in conjunction with the strips of dust on top of the piping. The heat radiates freely from the clean sides and the bottom of the pipes, but is retarded or stopped altogether from flowing on the top, where the dust is. This goes on for years and forms a species of electric couple. I notice the corrosion is right under where the dust would accumulate. Some years ago I had a case of grooved pipes from somewhat the same cause.

W. A. Cooil.

Ladysmith, B. C.

Piston Rod Cracks Cylinder

An unusual mishap occurred recently to a 150-hp. medium-speed slide-valve engine in a small manufacturing plant. The piston rod of this engine is threaded into the crosshead and is secured with a jam nut. One day the jam nut worked loose, and as the threaded shank of the rod was a rather loose fit in the crosshead boss, the rod started to back out gradually, and in a little while had worked toward the head end a distance equal to the normal clearance between the piston and the cylinder head. The engine then emitted a gentle pound, which quickly increased in volume until

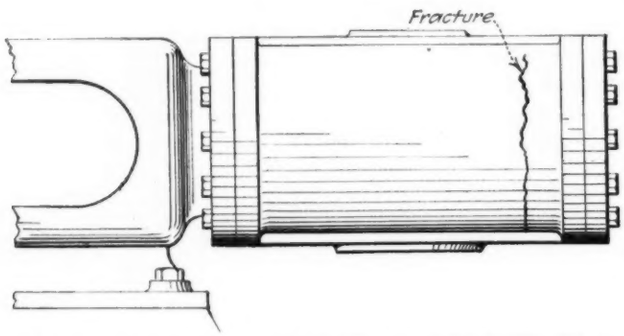


FIG. 1. LOCATION OF FRACTURE IN CYLINDER WALL

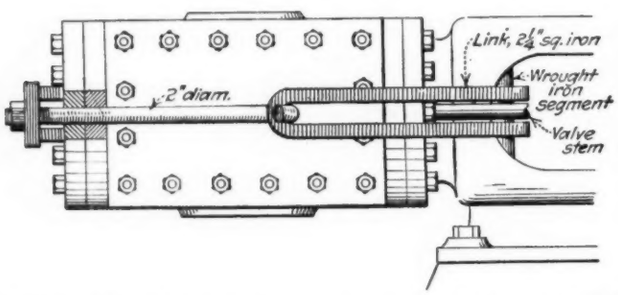


FIG. 2. SHACKLE ON STEAM-CHEST SIDE OF CYLINDER

it sounded like the rhythmic thump of a steam hammer. The engineer, who was standing some distance away, immediately rushed to close the throttle. But before the engine could be brought to a standstill, the piston had advanced far enough to strike the cylinder head an extra hard lick, and the damage was done.

Ordinarily, in an accident of this kind the cylinder head would have been shattered and the studs would have been stripped or broken, but in this case the wall of the cylinder was cracked along an arc that extended exactly parallel with the flange, as indicated in Fig. 1.

An examination of the iron showed it to be of a spongy and seamy texture for about one-quarter of an inch on each side of the crack, but in all other parts

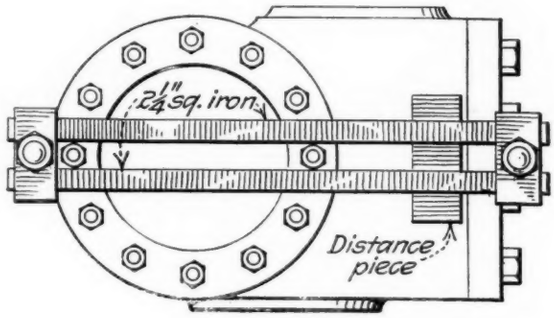


FIG. 3 SHACKLE BAR ACROSS CYLINDER HEAD

of the cylinder the metal showed a close-grained texture. The flaw was apparently due to the localization of an admixture of impure iron in a narrow streak roundabout the casting.

The damage was repaired by shackling the head end of the cylinder to the barrel of the frame, as indicated in Figs. 2 and 3, after the fracture had been filled with a mixture of litharge and glycerin. On the steam-chest side a wrought-iron segment was fitted to the curved end of the opening in the barrel, so as to provide a flat surface to pull against.

A. J. Dixon.

St. Louis, Mo.

Incorrect Steam Pressure Caused Reduced Vacuum

Our 6,000-kw. turbine has a steam jet-vacuum pump, the steam to which is supplied through a reducing valve which keeps the pressure always at a certain value. Normal vacuum is 28.75 in., but recently it dropped to 28.65 in. and it was very much of a puzzle to discover the reason.

The amount of steam to the steam seals was both increased and decreased without effect. The casing bolts and joints were tightened and painted, but no improvement. We checked the vacuum gage and found it in good condition, repacked all the valve stems in the vacuum line and shellacked them over, but that did not help any. The unit was shut down and the condenser filled with water, but not the slightest leak was found. The condenser temperatures were normal, which indicated proper operation. This narrowed the trouble down to the air pump, which we dismantled and found the jets to be clean and not stopped up and the whole pump in perfect operating condition. Watching the reducing valve showed that it was keeping the steam constant at 135 lb., which had been found to be the best value. Changing this pressure was tried, and it was found that the results were inconsistent with normal operation. The steam gage was then tested and found to be reading 10 lb. low. When a corrected gage was installed and the pressure set at 135 lb., the vacuum immediately increased to its normal value and an appreciable improvement in economy resulted.

Wichita Falls, Tex. Leslie R. Gray.

Boilermakers as Inspectors

Like A. D. Palmer (page 714, *Power*, May 3) I have often wondered why boilermakers have not become boiler inspectors and qualified to take the lead in this line that by logical reasons should be theirs.

While these men can make substantial repairs on boilers, knowing how and why they should be done, yet the minute you begin to talk "mathematics of the boiler" to them, they display the disposition that Mr. Palmer has described. I have reasoned and pleaded with them to take this matter up, but all to no avail, and yet they will criticize and find fault with the inspectors and want to know what right they have to "butt in" on the boilermaker's work.

It is hard to get the rank and file of boilermakers interested in periodicals devoted to their special field, but I hope that some day they will see the folly of their way and take up the mathematics of the boiler, and then I believe they will do as well in this as they have in the physical branch, for as mechanics they are as near ambidextrous as it is possible to be.

Cambridge, Mass. W. K. CAMPBELL.

Elongation of Cylinder-Head Bolts

A number of years ago there was much discussion as to the question of the amount of tightening-up tension necessary to give cylinder-head bolts to avoid leakage at the point when steam was turned on. In Fig. 1 *A* this problem is illustrated. If the joint between the cover *A* and the cylinder barrel *B* is compressed by an initial tension T_0 being placed on the bolt then it has been agreed that if the total steam pressure *Q* acting on the head does not exceed the value of T_0, the joint of *A* and *B* will not separate. This is evidently true, and acting on this it is generally believed that as long as *Q* does not exceed T_0 no additional load will be placed on the bolt, consequently there will be no elongation of the bolt until *Q* exceeds T_0. If a false head *D*, Fig. 1 *B*, is added the question arises, Are the conditions changed, causing the pressure *Q* to place an additional strain on the

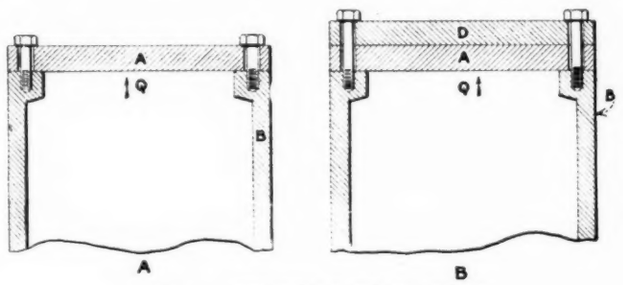


FIG. 1. CYLINDER-HEAD JOINT

bolts and producing a further elongation? On the surface, at least, it would seem that the false head *D* should have no effect, being about the same as a washer under the bolt head.

The same load condition of Fig. 1 exists in the crank end of a connecting rod such as shown in Fig. 2 *A*. It is assumed by most engineers that as long as the pressure *Q*, such as due to enertia of the moving piston, does not exceed the initial tension T_0, no separation of the two bearing halves will occur and no elongation of the bolts takes place.

However, if the bearing be of the type shown in Fig. 2 *B*, then any pressure *Q* such as inertia stresses will produce an additional elongation of the bolts and at the same time will decrease the pressure between the joints. This has been proved by a series of tests in England, where an elongation was always observed with an increase in pressure in *Q;* however, the decrease in the joint pressure was not equal to the increase in loading *Q*. If these tests be accurate, it would seem that with Fig. 2 *A* a similar elongation would take place,

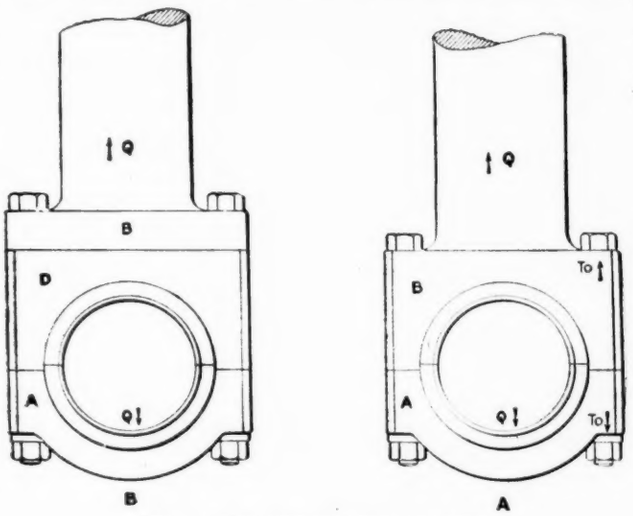


FIG. 2. ENGINE CRANK BEARING

and it may be an open question as to whether, in Fig. 1, an elongation is impossible as long as the steam pressure is less than the initial bolt tension. G. GROW.

Philadelphia, Pa.

Clinker Crystallizing Kink

The communication in the issue of *Power* of April 26, 1921, page 676, "Clinker Crystallizing Kink," should be given careful consideration before an installation of this kind is attempted. The article presents only one side of the case, and as it gives the good points only of a piece of apparatus, it is incomplete, and in the attempt to adapt this equipment to any particular installation, some engineer will probably run into a lot of expense.

In the use of steam, to quote Mr. Baker, "It crystallizes the clinker, and it takes less exertion on the part of the men to break it off the brickwork." Now what happens when the firemen is informed regarding this and observes the results and the amount of work that has been saved? Naturally, the steam valve is opened, and if a little is good maybe a lot is better. Assume that a drilled nipple or a piece of double-extra strong pipe is inserted in the line so that a full valve opening will allow only an equivalent area of 0.09 sq.in. Let us figure what this is costing. At 200-lb. pressure we will have about 1,000 lb. of steam per hour. Of course there will be two pipes, one on each side wall. This gives 2,000 lb. per hour, which, at an estimated cost of 50c. per 1,000 lb. is equivalent to $1 per hour.

There is a further loss due to the admission of this relatively cold gas into the combustion chamber, all the steam so admitted has to be heated to the temperature of the gas in the furnace.

In using exhaust steam for this purpose, we require larger pipes and have the loss mentioned in the preceding paragraph; also, exhaust steam is not always a free commodity and should be put to more efficient use than treating clinkers.

This should not be considered as a mere theoretical criticism of Mr. Baker's article, but from actual experience with a similar idea under the most careful watching of the steam valves, it was found to be too expensive and was taken out. Accurate daily records of the ratio of net steam output to gross steam output will quickly show what is happening.

There are other means of overcoming this clinker zone trouble, which are much more efficient and involve only a small annual fixed charge against the boiler room.

H. E. MITCHELL.

Erie, Pa.

Burning Powdered Coal

The article by F. R. Low, page 745 of the May 10 issue, explaining the necessity for a large combustion chamber for burning coal, was interesting.

If it were possible to carry a pressure or a vacuum in the combustion chamber, it would not be necessary to build them so large. When boilers were first operated with hand firing and natural draft, small combustion chambers, as we understand them now, were the rule, and a slight vacuum was present in the furnace. As soon as it was realized that higher boiler ratings could be obtained by passing a larger volume of gas over the heating surface, it became necessary to make use of the stokers and forced draft in order to obtain this result.

It was then found that as soon as pressure was carried in the furnace, the refractory substances composing the lining of the furnace were rapidly destroyed, and that a balanced condition must be maintained with no pressure in the furnace if low maintenance cost was to be expected.

The early experiments with powdered coal under boilers failed because all the air for combustion was supplied at the burner and small combustion chambers were used, and for several years it was maintained that powdered fuel could not be burned under boilers until a better refractory substance was secured.

The present installations are meeting with success because they admit the air at various points of the furnace, so as to keep the temperature of the brickwork at a low point, and even then it is found necessary to admit 25 to 30 per cent excess air, in order not to have an excessive bill for maintenance of the lining of the combustion chamber.

It is possible to burn coal on modern stokers with as little excess air as 25 to 30 per cent, so that the main saving derived from the use of powdered coal is the elimination of the loss caused by the combustible in the ash, and the heat units and time lost for cleaning fires.

On the steam end of the power plant, practice has gone to higher and higher steam temperatures to obtain high efficiencies, and in the boiler room it is necessary to go to higher combustion temperatures if higher efficiencies are to be secured.

At combustion temperatures of 2,500 to 2,600 deg. and exit temperatures of 500 deg., an over-all efficiency of approximately 76 per cent is possible; whereas, if it were possible to approach open-hearth temperatures of 3,300 to 3,400 deg. in the combustion chamber with the same exit temperature, efficiencies of 83 to 85 per cent would be possible.

If furnaces could be operated with a slight pressure in the combustion chamber, it would be easy to obtain these temperatures, and the present large combustion chambers would not be necessary, but before any such operation can be expected it is necessary to find some refractory for furnace linings that will give much better results than can be obtained from any that are now on the market.

It seems to me that it would be possible for the various interested parties to get together and endow research work along this line in some of the research laboratories in this country.

J. B. CRANE.

Pittsburgh, Pa.

Are Small Boiler Tubes Desirable?

In the issue of Jan. 25, N. S. Riggin states that small tubes are a detriment to the efficient operation of the boilers. I wish to submit data procured on a test of two horizontal return-tubular boilers. This test was not taken for the purpose of determining what benefits could be derived by having large or small tubes, but to ascertain if these boilers would be ample to take care of an additional unit and a heating system to be installed. The tests were of six days' duration of ten hours a day. This was necessary in order to procure accurate data of the different conditions under different loads. During the test the furnaces and boilers were operated under every-day conditions except for the water and coal being weighed and measured as is usual.

DIMENSIONS AND DATA OF NO. 1 BOILER

Grate surface, sq.ft	30
Diameter, in	72
Length, ft	18
Number of tubes	70
Size of tubes, in	4
Heat units in one pound of dry coal, B.t.u	13,000
Steam pressure, gage, lb	130
Temperature of water fed to boiler, deg. F	210
Coal fired, lb	51,105.6
Water fed to boiler, lb	347,760
Equivalent evap. from and at 212 deg. F., lb	364,104.720
Equivalent evap. from and at 212 deg. F. per lb. of coal, lb	7.12
Combined efficiency, per cent	53.10

DIMENSIONS AND DATA OF NO. 2 BOILER

Grate surface, sq ft	30
Diameter, in	72
Length, ft	18
Number of tubes	86
Size of tubes, in	3½
Heat units in one pound of dry coal, B.t.u	13,000
Steam pressure by gage, lb	130
Temperature of water fed to boiler, deg. F	210
Coal fired, lb	43,540
Water fed to boiler, lb	348,320
Equivalent evap. from and at 212 deg. F., lb	364,691.04
Equivalent evap. from and at 212 deg. F., per lb. of coal, lb	8.37
Combined efficiency, per cent	62.47

These two boilers are connected to the one chimney by a breeching in which a separate damper is placed for each boiler, and are controlled by a hydraulic damper regulator. The average draft over the fires was 0.22 in. of water. The furnaces are hand-fired and have shaking grates; the fires were cleaned and tubes blown on both boilers at 7 a.m. and 1 p.m. each day. As these two boilers are not large enough to handle the full load of the plant at one time, it was necessary to put on as many units each day as the boilers could handle. At no time did the No. 1 boiler evaporate more water than No. 2, but No. 2 boiler showed 9.37 per cent more efficiency and 7,565.6 lb. less coal than No. 1 boiler.

C. J. MILLER.

North Tonawanda, N. Y.

INQUIRIES OF GENERAL INTEREST

Re-Rolling Tube Ends After Beading—In retubing a horizontal return-tubular boiler should the tubes be re-rolled after beading if not found to leak after filling the boiler with cold water, and is such re-rolling objectionable? F. R.

If the tube joints do not leak after applying a test pressure of 50 per cent more than the working pressure then re-rolling would be of no benefit; but on the other hand, unnecessary rolling is detrimental to the structure and strength of the tube material.

Heat Loss from Bare Steam Pipe—What is the rule for rate of loss of heat from bare steam pipes exposed to the atmosphere? T. G. R.

The rate of loss of heat by radiation from bare iron steam pipes to atmospheric air per square foot of external pipe surface depends on the difference of temperature of the contained steam and that of the surrounding air, and is increased by the roughness of the exterior pipe surface and also increases with increase of the temperature difference.

For iron pipe of average roughness of surface in quiet room atmosphere, the heat loss in B.t.u. per square foot of pipe surface for a difference of temperature inside and outside of the pipe, up to 50 deg. F. difference is about 2 B.t.u. per hour per degree difference; for 50 to 100 deg. difference, about 2.2 B.t.u.; 100 to 200 deg. difference, about 2.7 B.t.u.; 200 to 300 deg. difference, about 3.3 B.t.u. 300 to 400 deg. difference, about 4 B.t.u.

Efficiency of Surface-Ignition Oil Engines—If a gas engine shows a falling off in efficiency on low loads, why does not a surface-ignition oil engine do likewise since it operates on the Otto cycle, as does the gas engine? P. L.

In a gas engine the mixture of fuel and air is compressed in the cylinder during the compression stroke. If the load is low, the weight of air and fuel mixture is less than at full load. The compression pressure will then be lower than with a heavy load. Since the efficiency of an Otto cycle engine is proportional to the ratio of the pressures at the beginning and end of compression, the efficiency drops at low loads; that is, low compression pressure.

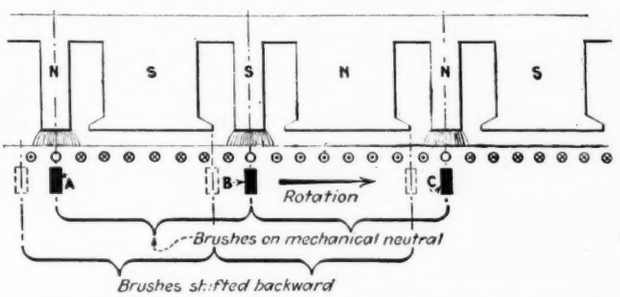
A surface-ignition oil engine on the other hand, compresses nothing but pure air, the fuel being injected at or about inner dead center. The compression of the pure-air charge is the same regardless of the amount of fuel ignited at dead center. The efficiency is practically constant at all loads.

Hydrostatic Test Pressure for Boilers—Why is the cold-water test pressure on a boiler 33 to 50 per cent more than the steam pressure at which the boiler is to operate? Is it to cover a margin of safety, or because there is less strength when the boiler is carrying steam pressure? D. J. S.

The pressure that a boiler can carry should be determined from computation, based on the design, workmanship and actual strength of the material. The safe working pressure is obtained by dividing this computed pressure by a factor of safety to cover imperfections of material, workmanship and estimates, and the condition, intended use and age of the boiler. A hydrostatic test pressure is employed to determine whether the joints probably would be tight and whether the strength of parts probably would become impaired from distortions when the boiler is operated under the estimated safe working pressure, and for the purpose the test pressure usually is 50 per cent greater than the predetermined safe working pressure.

Effect of Interpoles on Speed of Motors—Why do interpoles tend to keep the speed of direct-current motors constant from no load to full load? J. H. B.

The interpoles should have no effect whatever on the speed of an interpole motor at different loads, if the brushes are set in the neutral position. This position should be carefully located before the machine leaves the manufacturer, and the brushes fixed so that they cannot be easily moved from this position. With the brushes on neutral the interpole flux divides equally on both sides of the coil under commutation. Referring to the figure, of all the conductors in series between one pair of brushes, a certain number are therefore cutting one-half the flux from a north interpole, while the same number of conductors are cutting the same amount of flux from a south interpole. The net counter-em.f. induced by the interpoles is therefore zero and they can have no effect on the speed.



EFFECT OF BRUSH SHIFTING IN INTERPOLE MOTORS

In cases of faulty design, however, when the interpole is too powerful and causes sparking, it may be necessary to shift the brushes slightly backward, in order to move the coil under commutation into a weaker portion of the interpole field. Considering in the figure the conductors between brushes *A* and *B*, traveling under a south main pole, it may be seen that shifting the brushes backward increases the amount of flux from a north interpole and decreases the amount of flux from a south interpole, that these conductors are cutting. The counter-em.f. induced by the north interpole flux is therefore greater than that induced by the south interpole flux, and the difference between the two will oppose the counter-em.f. induced by the main south pole. The same thing will happen under each pole. Depending on the amount of brush shift, part or all of the counter-em.f. induced by the interpoles will therefore oppose that induced by the main poles, and if the net interpole counter-em.f. happens to be exactly equal to the resistance drop through the motor, its speed will remain constant at all loads. If the interpole em.f. is greater than the IR drop, the speed will increase with the load.

Boiler Tests with Pulverized Illinois Coal*

By HENRY KREISINGER† and JOHN BLIZARD‡

THIS paper gives the summary of the results of a series of eleven tests made by the Fuel Section of the United States Bureau of Mines in co-operation with the Research Department of the Combustion Engineering Corporation on an Edge Moor boiler fired with pulverized coal at the Oneida Street Station of the Milwaukee Electric Railway and Light Co., Milwaukee, Wis. The object of the tests was to determine what over-all efficiency can be obtained with pulverized Illinois coal under various conditions of furnace operation and different preparation of coal as to degree of fineness and percentage of moisture.

The tests were made in a thorough manner, everything being done to make the results accurate and reliable. The pulverized coal was weighed in specially designed tanks placed on platform scales as it was supplied to the furnace. The tests were from 17 to 25 hours in duration.

Tests Nos. 28 to 30, inclusive, were made with the usual preparation of coal as it is burned in the plant under ordinary operating conditions. Test No. 31 was made with the same condition of coal as the three previous tests, but with the furnace provided with a cooling coil over the hearth and along the walls near the bottom of the furnace to facilitate the removal of ash. Tests Nos. 32 to 35, inclusive, were made with the same furnace arrangement as test No. 31, but with the coal pulverized to a lesser degree of fineness.

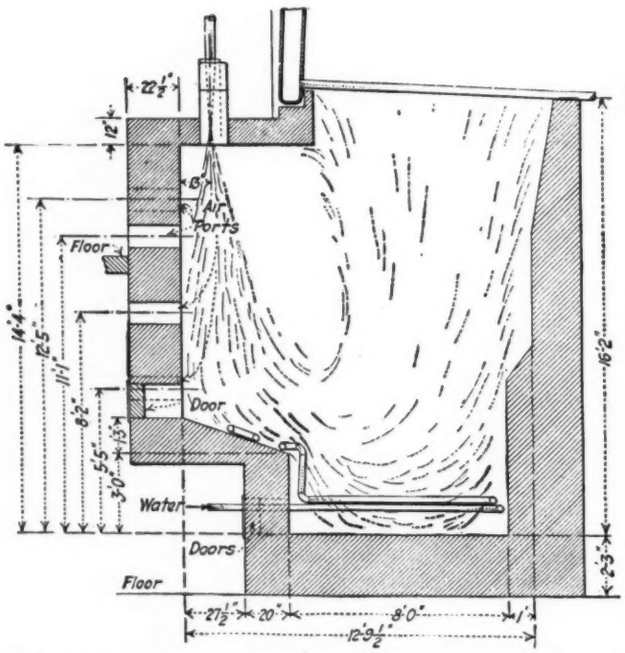


FIG. 1. SECTION THROUGH FURNACE, SHOWING ARRANGEMENT OF BURNER AND COOLING COIL

Tests Nos. 36 to 38, inclusive, were made with the same furnace arrangement as in the previous four tests, but with undried coal.

Fig. 1 shows the arrangement of the burner and the cooling coil over the hearth and near the bottom of the furnace. The cooling coil consisted of three lengths of 2-in. pipe over the hearth and two along the side and rear walls, the total surface being 48 sq.ft.

Fig. 2 shows the coal-weighing apparatus, which was placed between the storage bin and the feed bin. There were two burners and two feeders, and the coal to each feeder was weighed separately. The weighing tanks were connected to the storage bins and the feeder bin by flexible canvas connections to permit weighing and to prevent the coal dust from escaping into the room when the weighing tanks filled and emptied. The tests were started and closed with the feeder bins empty.

The feed water was weighed in two water tanks placed on platform scales. The water supplied to the cooling

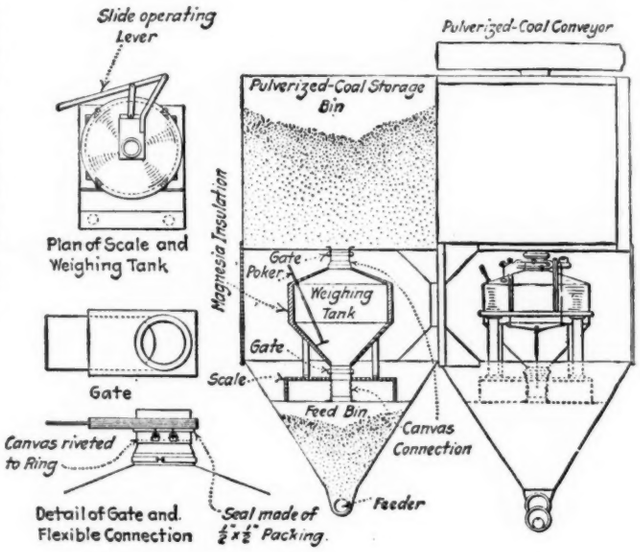


FIG. 2. COAL-WEIGHING APPARATUS USED IN TESTS

coil was measured by a 2-in. meter which was calibrated at the rate of feeding the water through the cooling coil, and its measurements were found reliable to within less than one-half of one per cent.

Flue-gas samples were taken at six points in the uptake and collected over one-hour periods. Flue-gas temperatures were measured with thermocouples at the same six points where samples were drawn for analysis, and readings were taken every 15 minutes. The flue-gas temperature given in Table I is the average of the measurements with the six couples.

Results of the Tests

The results of the tests are given in Table I. The quantities of heat absorbed by the boiler, superheater and cooling coil, when the latter was used, are itemized separately. In the heat balance the losses by radiation are given by a separate item. In a series of tests on the same boiler and setting the radiation loss per square foot of exposed surface should be nearly constant and should vary only slightly with the capacity developed by the boiler. For the calculation of the radiation loss it was estimated that 250 B.t.u. were lost per square foot of the exposed surface per hour when the boiler was operated at 100 per cent of rating, and 350 B.t.u. when operated at 200 per cent of rating. The radiation loss was calculated according to the percentage of rating developed. These calculations of the radiation loss leave the true "unaccounted for," which consists of errors. In a series of well-conducted boiler tests this true "unaccounted for" should be close to zero and should vary on both sides of the zero line according to whether the plus or minus errors predominate.

It has been assumed that in order to get good results the coal must be pulverized to a fineness of 95 per cent through

*Abstract of paper presented at Chicago, Ill., May 24, 1921, before the American Society of Mechanical Engineers.

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‡Fuel Engineer, United States Bureau of Mines, Pittsburgh, Pa.

a 100-mesh screen and 85 per cent through a 200-mesh screen. Table II gives the results of complete sizing tests of the coal burned in Tests Nos. 32 to 35, inclusive. The coal was much coarser than specified by the foregoing statement. The results of these tests seem to indicate that it is not necessary to pulverize the coal to the extreme fineness of 85 per cent through a 200-mesh screen in order to get good combustion and good efficiency. The completeness of combustion seems to be more a matter of a proper furnace and burner design and the right way of supplying air than of the fineness of the coal. The losses due to coarseness of coal would be shown by the greater percentage of carbon in the refuse. The average loss due to this cause for the four tests with the coarser coal is 0.7 per cent. The average of this loss for the previous four tests is 0.6 per cent. The averages of the efficiencies are very nearly the same.

The ability to burn coarser coal means increased capacity of the pulverizing mills and decreased cost of coal preparation.

Another assumption that has been generally accepted is that coal must be dried to about 1 per cent moisture in order to be successfully burned in pulverized form. In order to determine to what extent this is true, tests Nos. 36, 37 and 38 were run with undried coal. The results show that the completeness of combustion was as good as with the dried coal. There was no loss due to CO in the flue gases and the losses due to combustible in the refuse averaged only 0.3 per cent for the three tests, which is in fact less than the average with the dried coal.

TABLE II. RESULTS OF SIZING TESTS OF COAL BURNED IN TESTS NOS. 32–35

Test No.	— Percentage of Coal Passing Through Screens — 20-mesh	40-mesh	100-mesh	200-mesh
32	99.9	99.2	93.2	67.0
33	99.9	99.2	93.1	70.1
34	100.0	98.9	90.8	65.5
35	99.8	98.0	88.6	64.0

The losses due to moisture in coal increased 0.5 to 0.6 per cent, which is at the rate of about 0.1 per cent for every 1 per cent of increase of moisture in the coal. The average decrease in the boiler efficiency for the three tests is about 0.7 per cent, which checks closely the increase in the losses due to increased moisture in the coal. It seems, therefore,

TABLE I. SUMMARY OF RESULTS OF STEAMING TESTS ON AN EDGE MOOR BOILER BURNING POWDERED COAL, AT ONEIDA STREET POWER STATION, MILWAUKEE, WIS.

Heating Surface
Boiler 4,680
Superheater 594
Furnace coil 48
Total ..... 5,322 sq.ft.

Draft—Natural
Burner—Lopulco vertical
Coal feed—Screw and air blast.

Volume of furnace ..... 1,660 cu.ft.
Greatest height of furnace ..... 16.7 ft.
Greatest width of furnace ..... 9.3 ft.
Greatest length of furnace ..... 13.3 ft.

1 Test number	28	29	30	31	32	33	34	35	36	37	38
2 Duration, hours	22.90	23.72	18.17	23.62	23.60	22.47	25.22	23.53	23.43	23.33	16.73
Coal as Fired:											
3 Per cent through 100 mesh	96.10	95.80	....	95.40	93.20	93.10	90.80	88.60	....	94.40	95.40
4 Moisture content, per cent	1.42	2.92	2.75	2.55	3.79	3.07	3.60	3.47	7.69	8.23	8.23
5 Volatile matter, per cent	36.62	36.66	37.45	36.58	36.57	36.29	37.17	36.27	35.82	34.42	34.70
6 Fixed carbon, per cent	48.16	46.63	46.08	48.07	48.43	49.01	46.39	48.87	45.74	46.39	44.67
7 Ash, per cent	13.80	13.79	13.72	12.80	11.21	11.63	12.84	11.39	10.75	10.96	12.40
8 Sulphur, per cent	2.66	3.64	3.49	2.92	2.40	2.66	2.43	2.99	2.10	2.21	2.90
9 Calorific value, B.t.u.	11,956	11,860	11,875	12,085	12,172	12,178	11,889	12,188	11,565	11,508	11,245
10 Total fuel fired, lb.	40,214	39,862	52,746	46,613	43,947	45,469	47,459	41,250	43,729	41,640	36,315
11 Coal fired per hour, lb.	1,756	1,681	2,903	1,973	1,862	2,024	1,882	1,753	1,866	1,785	2,171
12 Coal fired per hour per cu.ft. of combustion space	1.10	1.05	1.81	1.23	1.16	1.26	1.18	1.09	1.16	1.12	1.36
Ash and refuse:											
13 Carbonaceous content in furnace slag, per cent	0	0	0	0	0	0	0	0	0	0	0
14 Carbonaceous content in 2d and 3d pass refuse, per cent	4.15	3.49	5.00	5.25	9.52	7.19	7.52	7.37	4.97	4.32	4.49
15 Carbonaceous content in uptake dust, per cent	4.95	5.24	7.35	5.13	7.70	6.45	7.81	5.57	3.91	3.50	3.28
16 Calculated total carbon in refuse, per cent of coal fired	0.50	0.54	0.62	0.36	0.87	0.61	0.67	0.30	0.24	0.22	0.26
17 Softening temperature of coal ash, deg. F.	.....	2,050	2,210	2,120	2,120	2,110	2,,060	.....	.....	.....	.....
Ash account (per cent of ash fired)											
18 From bottom of furnace	29.20	25.50	41.50	48.10	10.10	24.10	37.80	59.40	48.10	47.10	41.60
19 From 2d and 3d pass	12.10	11.60	5.80	10.40	10.30	7.00	9.10	10.50	8.30	7.70	8.70
20 From dust collector	31.50	25.00	33.20	29.20	27.50	29.00	25.50	30.10	32.80	26.00	26.30
21 Unaccounted for	28.20	37.90	19.50	12.30	52.10	39.90	27.60	0.00	10.80	19.20	23.40
Air:											
22 Temperature, air at furnace, deg. F	83	90	80	76	85	71	64	74	79	72	75
23 Pressure air at feeder, inches of water	5.00	5.30	7.50	5.10	5.60	5.50	6.40	5.90	5.70	5.50	6.20
24 Excess air in flue gas, per cent	30	22	18	18	19	15	20	25	19	16	20
Flue gas:											
25 Carbon dioxide, per cent by volume	14.10	14.90	15.40	15.50	15.30	15.80	15.10	14.60	15.50	15.80	15.40
26 Oxygen, per cent by volume	4.80	3.80	2.90	3.30	3.20	2.40	3.60	4.10	3.20	2.90	3.60
27 Carbon monoxide, per cent by volume	0	0	0.26	0	0	0.10	0.10	0	0	0	0
28 Pounds of dry flue gas per pound of coal	12.40	11.20	10.60	11.00	11.30	10.80	11.00	11.70	10.70	10.40	10.60
29 Temperature of gases in uptake, deg. F	517	492	610	483	457	472	470	486	484	466	514
Draft:											
30 At uptake, inches of water	0.12	0.10	0.27	0.09	0.06	0.08	0.05	0.09	0.10	0.05	0.15
31 Top of furnace, inches of water	0.00	0.02	0.00	0.00	0.01	0.00	0.01	0.02	0.02	0.03	0.03
Steam and water:											
32 Steam pressure, lb. absolute	184	186	196	189	188	187	185	186	183	182	185
33 Degrees superheat	60	59	80	58	53	61	67	79	74	70	102
34 Temperature of feed water, deg. F	108	99	99	101	101	100	100	103	98	100	99
35 Temperature of water to coil, deg. F		No coil		54	53	52	50	48	46	46	46
36 Temperature of water from coil, deg. F		No coil		129	146	145	118	151	139	130	144
Rates of heat absorption:											
37 Per cent of builder's rating (boiler only)	106.6	103.9	167.4	111.7	102.7	112.4	103.4	104.8	101.9	94.4	113.5
B.t.u. absorbed per sq.ft. of h. s. per hour:											
42 By water in boiler	3,567	3,482	5,575	3,743	3,437	3,758	3,466	3,508	3,413	3,207	3,798
43 By steam in superheater	892	861	1,856	920	772	845	869	1,155	1,039	934	1,573
44 By water in furnace coil		No coil		33,900	37,600	40,770	31,270	33,190	21,920	20,160	18,440
Heat absorbed per pound of coal as fired, B.t.u.											
45 By water in boiler	9,500	9,690	8,990	8,870	8,630	8,710	8,625	9,360	8,560	8,405	8,190
46 By steam in superheater	301	304	380	277	246	249	274	391	330	311	430
47 By water in coil		No coil		843	995	988	815	335	575	554	414
48 Total absorbed	9,801	9,994	9,370	9,990	9,871	9,947	9,714	10,086	9,465	9,270	9,034
Heat Balance (per cent of heat in coal fired)											
Heat absorbed:											
49 By water in boiler	79.4	81.6	75.6	73.4	70.8	71.5	72.4	76.8	74.0	73.0	72.7
50 By steam in superheater	2.5	2.5	3.2	2.3	2.0	2.0	2.3	3.2	2.8	2.7	3.8
51 By water in coil		No coil		7.0	8.2	8.1	6.9	2.7	5.0	4.8	3.7
52 Total and thermal efficiency	81.9	84.1	78.8	82.7	81.0	81.6	81.6	82.7	81.8	80.5	80.2
Heat carried away:											
53 By dry gases	10.8	9.1	11.4	8.9	8.2	8.6	9.0	9.5	9.0	8.5	9.9
54 By steam from burning hydrogen	4.1	4.3	4.2	4.1	4.1	4.2	4.2	4.0	4.2	4.1	4.2
55 By steam from moisture in coal	0.1	0.3	0.3	0.2	0.2	0.3	0.3	0.3	0.8	0.9	0.9
56 By steam entering with air	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
57 By carbon monoxide	0.0	0.0	1.0	0.1	0.1	0.2	0.4	0.0	0.0	0.0	0.0
58 By carbon in ash and flue dust	0.6	0.6	0.7	0.4	1.0	0.7	0.8	0.4	0.3	0.3	0.3
59 By radiation	2.5	2.6	1.9	2.2	2.3	2.1	2.3	2.4	2.4	2.5	2.2
60 Heat unaccounted for	—0.3	—1.3	1.6	1.3	2.9	2.2	1.3	0.6	1.4	3.1	2.2
61 Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

* Cooling coil in operation during first 8½ hours of test only.

that it is not necessary to dry coal down to 1 per cent of moisture in order to get good boiler efficiency; in fact, it seems that most of the Eastern coals can be pulverized and burned with good results without drying.

Capacity of Boiler That Can Be Developed with Pulverized Coal

The capacity of boiler that can be developed with pulverized coal depends entirely upon the size and shape of the furnace. With the present knowledge of the art of burning powdered coal the best results are obtained when the coal is burned at the rate of 1 to 1½ lb. per cu.ft. of combustion space per hour. Good results can be obtained when the coal is burned at rates varying from ½ to 2 lb. per cu.ft. of combustion space per hour, which gives a considerable working range. In Table I the rate of combustion is given by item 12. The range covered by this series of tests is from 1.05 to 1.81 lb. of coal per cubic foot of combustion space. If it is desired to operate the boiler at high rates of working, a large furnace must be installed, and the combustion space must be so arranged that the flames are given the longest possible path through the furnace. The design of burners and the admission of air are very important at high rates of combustion. It appears probable that future developments in the design of furnaces, burners and the air supply may make possible higher rates of combustion than the limit given herein.

Production of Electric Power by Public-Utility Plants

A comparison between public-utility power plants of the various states on the basis of their production of electricity and consumption of fuels during 1920, together with other pertinent data, is given in the latest bulletin issued by the Geological Survey.

The comparison between the states is given in the accompanying table. The first column shows the proportion of the country's power produced in each state; it may be seen that 64 per cent of the total was produced by ten states—New York, Pennsylvania, California, Illinois, Ohio, Michigan, Massachusetts, Washington, Montana and West Virginia. The population of these states, on the other hand, is but 42 per cent of the total population of the United States.

The second and third columns show the proportions of each state's electrical production attributable to water power and to fuel power. The last three columns show the proportion of the total consumption of coal, oil and gas in the United States attributable to the production of electric power in each state.

The actual figures in kilowatt-hours from which these ratios were calculated have been omitted from the table for the sake of brevity, but any or all of them may readily be obtained by computation from the totals given here.

The total year's production of electricity for the United States was 44,003,977,000 kw.-hr., of which 16,544,696,000 kw.-hr., or 37.6 per cent, was from water power and 27,459,281,000 kw.-hr., or 62.4 per cent, was from fuel power. Fuel consumption included 37,243,612 tons of coal, 13,122,704 bbl. of fuel oil and 24,702,107,000 cu.ft. of gas.

The increase of the total power produced in 1920 over that for 1919 was 13.1 per cent; water power and fuel power increased in practically the same proportion. The consumption of coal for 1920 increased 6.1 per cent over that of 1919, while the consumption of fuel oil increased 18.2 per cent and of gas 15.4 per cent.

If the fuel oil and gas consumed in producing electricity as shown here be reduced to equivalent coal, the fuel oil and gas in 1919 would be 7.3 and 1.7 per cent, respectively, of the coal consumed in that year by public-utility power plants; and in 1920 the percentages would be 8.2 and 1.9 respectively. These figures indicate that it would require only about 10 per cent of the coal consumed each year by public utility power plants in 1919 and 1920 to replace the fuel oil and gas used in generating electricity.

Wood was used as a fuel for producing a total of 156,506,000 kw.-hr. in 18 states. This was done most extensively in Oregon, as that state generated 99,972,000 kw.-hr., or about two-thirds of the total "wood-power" output.

It should be remembered that these figures do not cover all the electric power produced in the United States, but only that of the public utilities; according to the bulletin, they "Were obtained from reports received from more than 3,000 electric power plants, including central stations, electric-railway plants, and certain other plants, each of which produced monthly for public use electricity amounting to 10,000 kw.-hr. or more. The capacity of the plants that submitted these reports is 90 to 95 per cent of the capacity of all the plants listed."

PRODUCTION OF ELECTRIC POWER AND CONSUMPTION OF FUELS BY PUBLIC UTILITY POWER PLANTS DURING 1921

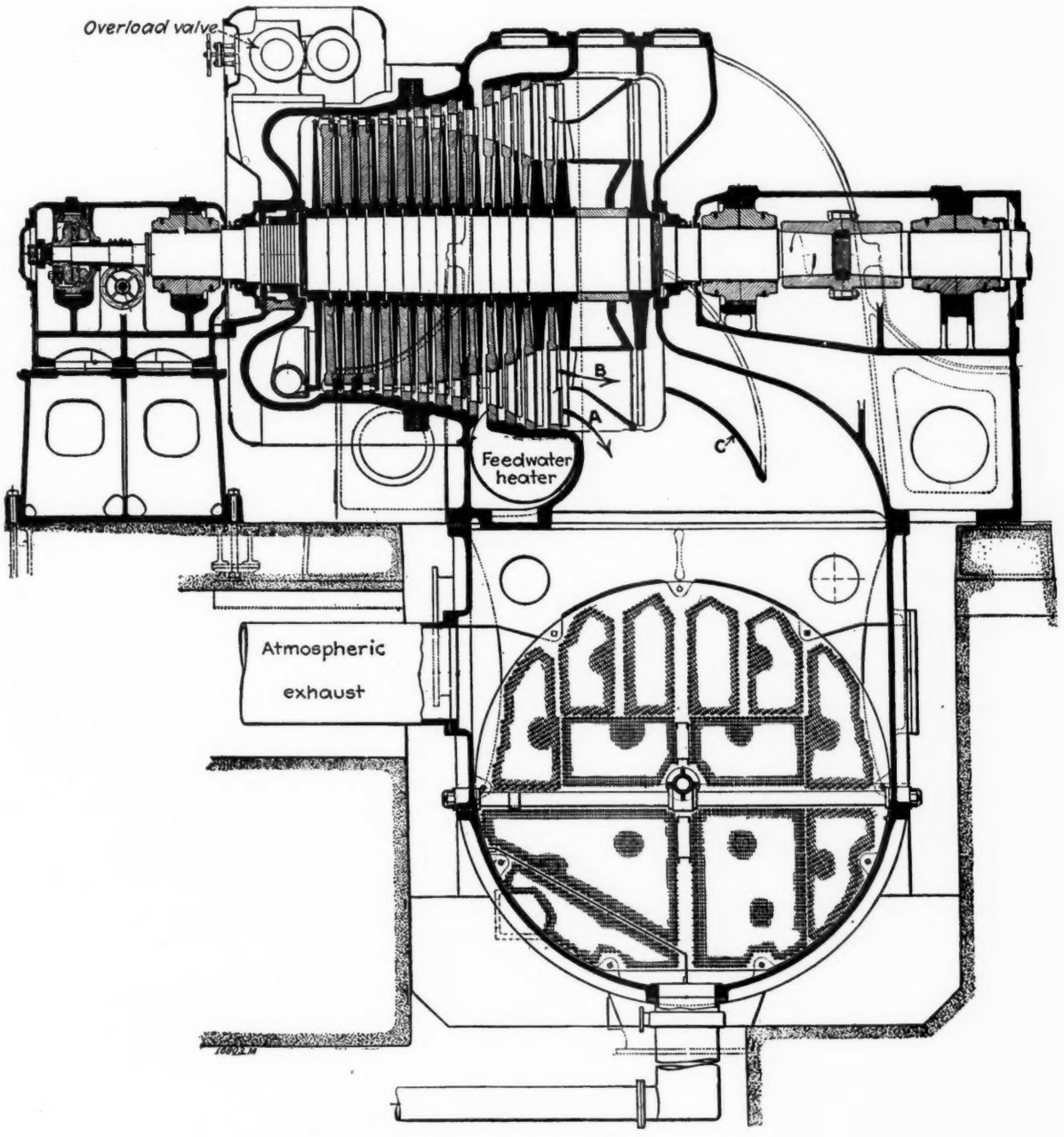
State	Per Cent of Country's Power Produced in Each State.	Per Cent of Each State's Power Produced by Water.	Per Cent of Each State's Power Produced by Fuels.	Per Cent of Country's Coal Burned in Each State.	Per Cent of Country's Fuel Oil Burned in Each State.	Per Cent of Country's Gas Burned in Each State.
Alabama	1.24	72.6	27.4	0.67	0.74	....
Arizona	0.36	66.9	33.1	0.01	1.82	10.93
Arkansas	0.27	1.2	98.8	0.33	0.64	12.04
California	8.45	69.1	30.9	0.00	43.02	....
Colorado	0.90	46.5	53.5	1.15	0.01	....
Connecticut	1.55	24.5	75.5	1.96	0.20	....
Delaware	0.17	0.0	100.0	0.32	0.00	....
Dist. Columbia	0.58	0.0	100.0	0.68	0.00	....
Florida	0.30	9.9	91.1	0.08	4.81	....
Georgia	1.35	82.5	17.5	0.33	2.38	....
Idaho	1.34	99.7	0.3	0.00	0.00	....
Illinois	6.82	6.1	93.9	11.82	0.03	0.10
Indiana	2.12	3.7	96.3	5.64	0.03	....
Iowa	2.30	65.2	34.8	2.69	0.07	6.00
Kansas	0.99	4.7	95.3	1.27	6.26	0.64
Kentucky	0.62	0.0	100.0	1.32	0.02	4.29
Louisiana	0.49	0.0	100.0	0.39	3.77	....
Maine	0.80	97.4	2.6	0.02	0.24	0.04
Maryland	0.64	1.0	99.0	0.98	0.00	....
Massachusetts	3.95	20.6	79.4	4.23	1.17	....
Michigan	4.75	32.1	67.9	5.00	0.01	....
Minnesota	1.59	58.4	41.6	1.40	0.04	....
Mississippi	0.15	0.0	100.0	0.38	0.84	....
Missouri	1.57	6.6	93.4	3.20	3.43	0.06
Montana	2.56	99.4	0.6	0.13	0.01	....
Nebraska	0.56	5.1	94.9	1.21	0.62	....
Nevada	0.07	97.4	2.6	0.01	0.02	....
New Hampshire	0.22	63.2	36.8	0.13	0.00	....
New Jersey	2.48	0.2	99.8	4.21	0.01	....
New Mexico	0.03	5.5	94.5	0.12	0.05	0.61
New York	15.54	42.8	57.2	12.77	0.30	....
North Carolina	1.86	85.4	14.6	0.64	0.01	....
North Dakota	0.06	0.0	100.0	0.48	0.01	19.95
Ohio	6.64	1.0	99.0	11.05	0.05	22.70
Oklahoma	0.48	0.9	99.1	0.27	2.64	....
Oregon	1.08	78.0	22.0	0.01	0.18	2.29
Pennsylvania	9.81	14.6	85.4	15.08	0.01	....
Rhode Island	0.76	1.8	98.2	0.78	2.46	....
South Carolina	1.65	92.2	7.8	0.32	0.00	....
South Dakota	0.01	30.4	69.6	0.22	0.18	....
Tennessee	1.30	77.0	23.0	0.77	0.00	5.72
Texas	1.54	0.6	99.4	0.64	22.07	....
Utah	0.48	100.0	0.0	0.00	0.00	....
Vermont	0.45	98.1	1.9	0.02	0.09	....
Virginia	1.18	38.9	61.1	1.22	0.02	....
Washington	2.72	96.5	3.5	0.12	1.07	14.40
West Virginia	2.53	2.1	97.9	3.26	0.00	....
Wisconsin	2.18	48.2	51.8	2.33	0.03	0.23
Wyoming	0.01	4.9	95.1	0.34	0.64	....

The largest system of automatic stations in the world will soon be put in operation in Kansas City, by the Kansas City Light and Power Co. The entire Edison system of the city is to be controlled from a series of five automatic substations, each containing two G.E. synchronous converters, operating independently. These five substations will be all fed from a central steam-generating plant, through a network of radial feeders. The capacity of the generating station at present is 60,000 kw. but the ultimate capacity is expected to be more than 150,000 kw. The Edison system will have an approximate capacity of 15,000 kw.

The Yosemite Power Co., of Groveland, Cal., has applied to the Federal Power Commission for permission to construct two reservoirs, a conduit and a power house on the South Fork of the Tuolumne River, a diversion dam near the mouth of South Fork and a seventeen-mile conduit to a second power house at Wards Ferry. Nine thousand horsepower is involved. Harry W. Deprez, of Shelbyville, Ind., has applied for a license covering power development at Government Dam No. 5 on the Muskingum River at Luke Chute, Ohio.

Rateau Turbine with Divided Exhaust

In a paper read before the Institution of Electrical Engineers in April, K. Bauman described an interesting turbine, the accompanying section of which is reproduced from *Engineering's* account of the meeting. It is of the Rateau type, steam, that coming from the outer portion of the last-stage blading is led directly to the condenser as shown at *A*, while that from the inner half of the blading is passed as shown at *B*, through another set of blading of the same length. The steam lines of the exhaust are controlled by guide-plates *C*, reducing the pressure drop between the last row of tur-



SECTION THROUGH TURBINE AND CONDENSER

was built by the Metropolitan Vickers Electrical Co. for the Dalmarnock Station of the Glasgow Corporation and has a maximum continuous rating of 18,750 kilowatts at 1,500 revolutions per minute.

The pipe type of nozzle box and steam chest is used, allowing the use of the highest steam pressures and temperatures without fear of distortion of the steam cylinder.

An integral feed-water heater is located as shown in the section, so as to abstract steam from one of the later stages while the vacuum is relatively low.

In order to utilize some of the residual velocity after it is no longer practical to use blades long enough to pass all the bine blades and the condenser and insuring the uniform distribution of steam over the condensing surface, reducing the pressure drop in the condenser itself. The importance of these is perhaps emphasized by the fact that the turbine overhangs so considerably the passage into the condenser.

Of the total force of 437 Patent Office examiners 110 have resigned in a little over one year, according to American Engineering Council. In the last year 142 of the 560 clerical workers left. The Council is trying to secure legislation to relieve this situation by allowing for larger salaries for Patent Office workers.

Explosions in Air Compressors*

MANY who have speculated on air-compressor explosions have attributed them to the ignition of oil vapor. The air in the compressor cylinder has a rather high temperature during a part of each cycle in consequence of the heat developed by compression. In an ordinary air compressor with tight valves the theoretical limit of temperature might be somewhere about 600 or 700 deg. F. if the compression took place without any loss of heat. Actually, however, there is a considerable cooling action on account of water jackets and intercoolers, so that the maximum temperature should not exceed something like 400 degrees.

It has been maintained that lubricating oil is likely to give off an inflammable vapor at this temperature, which, when mixed with the air in the compressor, is capable of exploding. Some of the opponents of this theory have, however, confidently stated that explosions occur in connection with high flash-point oils even *oftener* than they do in connection with those having low flash points.

Effect of Cylinder Wall Temperature

Most of the oil that is present in a compressor cylinder is lodged on the walls of the cylinder, and owing to the proximity of the water in the cooling jackets these walls do not have a temperature anywhere near as high as the air in the middle of the cylinder. Under good operating conditions and with well-designed apparatus the inner surface of the cylinder wall is probably not much more than 50 deg. F. hotter than the maximum temperature of the jacket water. If this is so it follows that the oil that is adhering to the cylinder is not giving off any significant amount of inflammable vapor, because the greater part of it is adhering to the comparatively cool metal. Some of the oil will be lodged on the face of the piston, and this may be hot enough to give off a certain amount of inflammable vapor if the flash point of the oil is lower than the maximum temperature of the air in the cylinder. It is important, however, to note that under normal working conditions the air in the cylinder is comparatively cool during the greater part of every cycle, the high temperature being attained only toward the end of the compression stroke. Hence, the total evaporation of oil from the faces of the piston will be far smaller than the advocates of the oil-vapor theory have probably assumed.

Oil *spray* in the cylinder or connections is sometimes suggested as a source of explosive action. It is hard to understand how spray could be formed from oil that has once adhered to the metal walls, except in consequence of the rupture of some part of the apparatus; any spray of this kind that does not immediately lodge on the walls is swept out into the receiver with the next discharge from the compressor; and moreover the amount of such oil spray would be so small that any explosion that it might produce would not even be noticeable.

Dust and "Carbon"

We come, now, to the consideration of such combustible *solid* matter as may be present in the cylinder or its connections; and here we are encountering a far more serious item. The free solid matter that is found in the cylinders, piping and receivers of air compressors is of two general kinds. First, it contains whatever may have been *introduced* into the air compressor in solid form; for example, dust drawn in through the intake. The dust held in suspension by the air that is taken into the compressor usually contains more or less mineral matter, which is likely to be either sandy or clay-like. Such matter is incombustible and probably would not be *directly* harmful except as it may increase the friction and the wear. But the incoming air may also contain combustible material, and this must be reckoned with. The combustible matter may be of an organic nature, or it may consist largely of particles of carbon. This last condition is most likely to prevail when the intake of the compressor is near a coal pile, or where it can take in more or less smoke from a chimney or stack.

"Carbonization" of the Oil

The other source of solid matter in the cylinder or receiver or connections is the so-called "carbonization" of the oil itself. We speak of solid deposits of this nature as "carbon," but although they contain free carbon they consist largely of oil in various stages of disintegration, and of compounds formed from the oil under the influence of heat and the oxygen of the air.[1]

Carbonization is due in large measure to the use of improper oil or of an excessive amount of it. The oil should have such a composition that it is reasonably stable at the temperatures that are likely to occur in the compressor or its connections; and this means, among other things, that it should not contain any animal or vegetable constituents, but should be purely mineral. Any oil will undergo a certain amount of decomposition, however, under the conditions to which it is exposed in air-compression apparatus. Slow oxidation is favored by the fact that the oil is spread out in a thin layer and exposed to heated air. Moreover, at the temperatures that prevail in the apparatus the oil is likely to "crack" to some extent. That is, it is likely to break down slowly with the formation of somewhat lighter oils and the simultaneous liberation of a certain amount of free carbon. No oil that is worth even momentary consideration contains, in its original state, any free carbon; and the percentage of carbon present in the oil (in combination with hydrogen) is no criterion whatsoever of the likelihood of the oil to deposit carbon. The carbon deposits thrown down by paraffin-base oils are said to be hard, gritty and specially adhesive, while those that are derived from asphaltic-base oils are described as fluffy and easily removable. However that may be it is certain that any oil that gives a considerable deposit of so-called carbon is objectionable in connection with air-compressor lubrication, whatever the nature of the deposit.

Ignition of the Carbon

It is well known that these "carbon" deposits take fire at times, and this can hardly be due to the air in the compressor attaining a temperature high enough to ignite pure carbon, especially when the carbon is in compact form.

It is not unlikely that the free carbon that is present in the deposits occasionally acts as a catalyzer, in promoting the rapid oxidation of the oily matter with which it is associated. Catalysis plays an important part in modern chemical manufacturing, and we are enabled by means of it to bring about many chemical reactions that will not occur unless a suitable catalytic agent is present. Charcoal and coke are among the substances used in this way to promote certain chemical actions; and perhaps it is not unreasonable to assume that the carbon in a compressor or its connections may sometimes induce rapid oxidation of the oily matter. It is known, too, that some of the products that would be formed in this way are themselves capable of acting as catalyzers, so that they might accelerate the action when it had once become well started. Fine metallic particles, worn off from the cylinder or the piston, are sometimes present in the deposit, and iron oxide often occurs to a significant extent, both being active catalyzers under certain conditions.

The heat generated by the beginnings of the oxidation would facilitate further action, and so the thing would go on until presently we might have a glowing or burning mass. This in itself might precipitate an explosion by heating the metal walls so that they could no longer withstand the normal air pressure to which they are subjected. Furthermore, the carbon deposit, if it happens to be of a light and fluffy nature, may be disturbed so that we have

*From an address by A. D. Risteen (Travelers Insurance Company) delivered before the Engineering Section of National Safety Council, at Philadelphia.

[1]Circular No. 99 of the Bureau of Standards, 1920, describes the nature of the deposit and the theory of its formation.

what is in effect a dust explosion, the finely divided carbon playing the part of the dust and the ignition taking place in consequence of the catalytic firing of the oily matter in the manner described.

Hard carbon deposits often prevent the valves of a compressor from closing properly, and this is likely to cause serious overheating. The piston, in making its return stroke to fill the cylinder with a fresh charge of air, will be followed by a certain amount of the compressed and therefore heated air, flowing back through the leaky valves from the receiver and its connections. Under these circumstances the contents of the cylinder are already, and before the compression begins, at a temperature considerably higher than that of the surrounding atmosphere; and this means that when the next compression stroke is completed the temperature in the cylinder may be a good deal higher than it would be if the valves were tight. The increase in temperature produced in this way and also by the friction of the air in churning back and forth through the restricted valve opening naturally accelerates any tendency toward the production of fire and is, therefore, a serious matter.

Removal of Carbon and Oil Deposits

Every effort should be made to prevent the accumulation of "carbon" deposits anywhere about the compression apparatus, and such matter as may be deposited in spite of all reasonable precautions should be cleaned away, frequently and regularly. A good deal can be accomplished in the way of cleaning the compression apparatus by discontinuing the oil from time to time and feeding soapy water into the cylinder for two or three hours—say once a week. The solution may be made up of one pound of soft soap to seven or eight quarts of water, and it is advisable to feed it into the cylinders at something like ten times the rate at which the oil is ordinarily supplied. The frequency with which this treatment should be given depends upon the experience with the particular air compressor under consideration. Some authorities recommend the introduction of a solution of lye into the pipe between the cylinder and the receiver every month or two. The lye solution may be made up by using one pound of lye to eight or ten quarts of water, and it may be delivered into the air discharge pipe by means of a sight-feed lubricator at the rate of 60 to 70 drops per minute. After using the lye or the soap solution the receiver should be blown off, and oil should again be used in the cylinder for a time before stopping the compressor. Otherwise rusting is likely to occur.

When hard carbon deposits form, which are not removable by the treatment here suggested, they should be cleaned out by whatever means may suggest itself as most effective. For instance, the air valves should be removed whenever necessary and cleaned by mechanical means. No kerosene or other inflammable solvent should be introduced with the air, however. This practice, which is far from unknown, is exceedingly dangerous.

Electrical Engineers To Hold Convention in Salt Lake City

The American Institute of Electrical Engineers will hold its 37th Annual and Pacific Coast Convention at Salt Lake City, June 21-24. A varied program of papers, speeches, trips and entertainment features has been arranged, including a welcoming address by the Governor of Utah. Technical sessions will be held during the morning of each day, the afternoons being reserved for excursions, golf, tennis, etc. A trip of especial interest will be made by automobile Thursday afternoon, June 23, to the Utah Power and Light Co.'s Terminal Station, which is the terminus of three 130,000-volt transmission lines and is said to be the largest outdoor substation in the world. Papers are to be presented at the technical sessions, beginning with Tuesday, as follows:

"Hydro-Electric Developments at Niagara Falls," by J. L. Harper, Niagara Falls Power Co.; "Modern Developments in Waterwheels," by W. M. White, Allis-Chalmers Co.

Wednesday, June 22, 9:30 A. M.—"Long Distance Transmission of Electric Energy," by L. E. Imlay, Superpower Survey; "Voltage and Power-Factor Control of 66,000-Volt Transmission Lines Connecting Two Generating Stations," by Raymond Bailey, Philadelphia Electric Co. Symposium on Long-Distance Transmission Systems: (a) "Voltage Regulation and Insulation for Large Power Long Distance Transmission Systems," by F. G. Baum, consulting hydroelectric engineer, San Francisco, Cal.; (b) "Some Transmission Line Tests," by W. W. Lewis, G. E. Co.; (c) "Notes on the Operation of Large Interconnected Systems," by L. L. Elden, Edison Electric Illuminating Co., Boston, Mass. "Modern Production of Suspension Insulators," by E. H. Fritz, Pittsburgh High-Voltage Insulator Co., and G. I. Gilchrest, Westinghouse Co.

Thursday, June 23, 9:30 A. M.—"Voltage and Current Harmonics Caused by Corona," by F. W. Peek, Jr., G. E. Co.; "A Solution of the Porcelain Insulator Problem," by E. E. F. Creighton, G. E. Co., and F. L. Hunt, Turners Falls Power & Electric Co.; "Transformers for Interconnecting High-Voltage Systems or for Feeding Synchronous Condensers from Tertiary Winding," by J. F. Peters and M. E. Skinner, both of Westinghouse Co.; "Electric Strength of Air under Continuous Potentials and as Influenced by Temperature," by J. B. Whitehead and F. W. Lee, both of Johns Hopkins University.

Friday, June 24, 9:30 A. M.—"Little Things from Little Places," by Guido Semenza, local honorary secretary of A. I. E. E. for Italy, consulting engineer, Milan, Italy; "Synchronous Motors for Ship Propulsion," by E. S. Henningsen, G. E. Co.; "Magnetic Properties of Compressed Powdered Iron," by Buckner Speed and G. W. Elmen, both of Western Electric Co.; "Heat Losses in Conductors in A-C Machines," by W. V. Lyon, Massachusetts Institute of Technology.

Opportunity Offered To Become Trade Instructors

Twenty-five scholarships are being offered by the University of the State of New York to qualified trade and technically trained individuals who want to become teachers. Those who are selected to hold the scholarship and who satisfactorily complete a course of training lasting one year will be licensed for life to teach their trade in the vocational schools of New York State. The salaries for such work now range from $1,800 to $3,500 per year.

Each holder of a scholarship will receive at least $1,000 for the period of one school year, and will be required to be in attendance for ten months in the Industrial Teacher Training Department of the State Normal School at Buffalo. Applicants must meet the following requirements: At least five years of successful all-round experience in work of not less than journeyman's grade in some trade, industrial or technical occupation; a good general education and ability to speak, read and write the English language; not less than 21 or more than 36 years of age on August 1 of the year in which the appointment is made; of good moral character and in possession of good health; a citizen of the United States and a resident for one year of the State of New York.

On the basis of the qualifications of the candidates a merit list will be set up for each trade, industrial or technical occupation. There will be no formal examination, but candidates will be rated by examining committees as to general education, practical experience, loyalty, moral character and physical fitness. Candidates may be required to appear before the examining committees for a personal interview.

The twenty-five scholarships will be awarded to qualified persons with all-round experience in the following occupations:

Electrical construction, repairing and operating; machine shop work; automobile repairing; machine drafting and designing; architectural drafting and designing; sheet metal working; printing, including presswork and composition; bricklaying; painting and decorating; carpentry.

Detailed information and blanks upon which application for scholarships should be made may be secured from the Director of Vocational and Extension Education, State Department of Education, Albany, N. Y.

Separately Fired Superheaters

In a paper on superheaters, which was quoted at some length in *Power* for April 5, 1921, Arthur D. Pratt, of the Babcock & Wilcox Company, stated that while they have not yet come into general use, particularly in this country, separately fired superheaters have a considerable field which will probably increase in extent in the near future.

The essential features of design of the separately fired type are in general the same as for the integral. The steam which ordinarily flows through a number of sections in series should be given as nearly as practicable a counterflow action with that of the hot gases.

Where it is known that the maximum temperature required is at or above the oxidization temperature of the metal, the superheater is best designed on the counterflow principle throughout, but with the section closest to the fire designed to superheat the steam from well below the oxidization temperature of the metal to the ultimate temperature desired, and should be so designed that this section may be readily renewed without interference with the remainder of the superheating surface. In some cases where the temperatures desired are so high that the superheater elements would oxidize at a rate requiring replacement too frequently for commercial operation, a protecting non-oxidizable coating may be used on the elements exposed to the highest temperatures.

Separately fired superheaters are not as efficient as integral superheaters, but have some advantages that in certain classes of work may more than offset this. The main advantage is the wide range of superheat possible, regardless of the amount of steam handled. While the superheat from a separately fired superheater may, by changing firing conditions, be widely varied regardless of the amount of steam, it is also possible through regulating these same firing conditions to keep a constant temperature of the steam, regardless of the amount.

Where Separately Fired Superheaters Are More Satisfactory Than Integral Apparatus

Though the field may not be broad, there are undoubtedly cases where the installation of apparatus of this class will give results more satisfactory than could be obtained with integral superheaters. Some of these may be cited as follows:

Where steam is conveyed through long pipe lines, it may be advantageous to use an independently fired superheater at the point where the steam is used. In some instances steam may be transmitted half a mile or more and the advantage of using an independently fired superheater in such cases for certain classes of work can be readily appreciated.

Changes in steam apparatus are constantly being made in existing plants. Frequently, prime movers are sufficiently antiquated to warrant the installation of new and more efficient turbines or engines requiring for the best efficiency superheated steam, while the boilers are or may be readily modified to become as efficient as the modern design. In plants of this character, particularly where the pipe lines are long, it may be cheaper to install a separately fired superheater in connection with the new prime movers than integral superheaters in the existing boilers.

Certain industrial processes require heat treatments such that the temperature required can be obtained only through the use of a very high superheat, and a separately fired superheater may give the results desired.

In steam laboratories it is frequently desirable to test power-plant apparatus over a range of a combination of steam pressure and temperature conditions, impossible to obtain with a boiler and integral superheater, and the separately fired superheater offers an ideal apparatus for this work.

In technical-school engineering laboratories some of the apparatus to be tested by students is not suited for superheated steam, while other apparatus requires it for satisfactory results, and a separately fired superheater meets these conditions.

Manufacturers of valves and fittings are called upon to test their product under conditions of temperature and pressure that cover a range difficult to obtain with an integral superheater, and for this purpose separately fired superheaters meet the requirements.

There is one field for the use of separately fired superheaters that, while not as yet open, may develop at some future time. In Great Britain and Europe there are a number of plants using what are termed "Re-superheaters." In these plants steam is taken from a turbine at a stage where the initial superheat has been practically lost and returned to secondary superheaters within the boiler setting, again superheated and returned to the next lower turbine stage in such condition. It would seem that if this practice ever becomes general and a separately fired superheater is developed that would give efficiencies approaching those of the integral superheater, a simpler arrangement could be worked out with such apparatus than is possible where the steam is returned to and divided among a number of integral re-superheaters.

Bureau of Standards Conducting Research on Refrigeration Constants

A recent bulletin from the United States Bureau of Standards describes the work that the Bureau has been doing for a number of years in the way of determining the thermal properties of material used in the production of artificial refrigeration. The following, which is quoted from the bulletin, indicates the scope of the project and the progress made so far:

This work has involved, first, the procuring of the fixed equipment required; second, the selection and training of a staff of investigators who were capable of carrying out the refined physical measurements; third, the development of methods of measurement suitable for the considerable range of temperatures and pressures involved and the construction, testing, and installation of the special apparatus required; fourth, making the measurements and preparing the results for publication in scientific papers, and making the results available for use by tabulating them in the forms required in engineering practice.

The work so far completed deals with the specific heats of brines, the specific and latent heats of ice, and the specific heat, latent heat and vapor pressure of ammonia, and with heat transmission through insulating materials and structures. The work completed on ammonia serves as the basis of a complete table of thermodynamic properties of the liquid and saturated vapor, for use in refrigerating engineering.[1] Work on the properties of superheated ammonia vapor is under way and should within the year yield data to complete the tables of properties of ammonia. These tables will be the equal, if not the superior, in point of accuracy, of existing data on the properties of steam. The methods and apparatus developed during the work on ammonia are available and should be used to measure the properties of other media now being used in refrigeration.

When we consider that existing tables are based largely on the work of Regnault done fifty or sixty years ago, it is obvious that systematic measurements of this kind should be made with modern facilities.

It should be pointed out that the work that has been completed thus far is of enduring scientific and engineering value. The best available modern instruments of precision measurement have been utilized, the accuracy that has been attained is high, and the results have been checked in every instance by independent methods of experimentation and often by entirely different experimental equipment and by different observers.

Of the several steps involved in such a program the first three enumerated are largely completed. The present program contemplates extending the work to the determination of the properties of materials other than ammonia, thus utilizing efficiently specialized experience and equipment which has been accumulated. Materials on which work is contemplated include carbon-dioxide, ethyl and methyl chlorides, and perhaps sulphur dioxides.

Pipes up to 96 in. diameter and 20-ft. lengths are now welded by the hammer process. This material is used for boiler shells, tanks, hydraulic and gas lines, penstocks and other apparatus where rivetless forged steel is desirable. The efficiency of the weld, based on several tests made by the National Tube Co., is 90 per cent of the strength of the material.

[1] This table appeared in *Power* for Jan. 18, 1921.

Washington News

The Trend in Coal Legislation—Muscle Shoals Project Discussed—Hoover Gets a Good Start—Work of the Power Commission

By Paul Wooton
Washington Correspondent

That the two Frelinghuysen coal bills will be enacted into law at this session of Congress is an opinion very generally held on Capitol Hill. It is believed that the seasonal rate bill will go through without any great amount of discussion. The fact-finding bill, however, is certain to draw some opposition. The prospect is that its legislative course will require more time than the other, and that before it is passed it will be simplified very materially. The question of just who is to do the fact finding is certain to be the matter of controversy.

If an effort is made to put through legislation that will mean the uprooting of the Geological Survey's division of coal statistics and its transplanting in the Department of Commerce, it will certainly result in the most extended discussion. Since Congress has authorized an intensive study of the reorganization of the Federal departments, it seems probable that the majority of Senators and Representatives would be inclined to vote againt any piecemeal shifting of activities from one bureau to another.

One of the underlying ideas of the fact-finding bill, it is believed, is to have data at hand so that Congress may have a better idea as to whether or not a strike is justified. This doubtless has a bearing on the hostility shown to this legislation by the United Mine Workers. While union labor has less influence on votes in Congress than ever before was the case, it still has some champions in each of the law-making bodies, who doubtless will erect some legislative obstacles in the way of the bill.

To Urge Work at Muscle Shoals

An investigation of the whole Muscle Shoals proposition has been made by a committee of the Farm Bureau Federation, which is the principal organization of farmers represented in Washington. The committee is now preparing its report and declines to comment on it in advance of its completion, but it is understood that it will urge strongly the resumption of work on the Wilson dam at Muscle Shoals at the earliest practicable moment.

Representative Graham, who, as the head of the committee which investigated war expenditures, showed particular enmity to the Muscle Shoals project, has introduced a bill authorizing the Secretary of War to lease the dam and nitrate plant No. 2. The Farm Bureau committee is said to be opposed to any such plan because of the difficulty likely to arise in arranging for the manufacture of cyanamid by other than the Government. Other objections are that Congress would demand such safeguards in connection with a lease as to make it commercially impracticable.

Hoover's Requests Allowed in Full

Quite contrary to the usual practice, the Committee on Appropriations of the House of Representatives acceded to the request of Secretary of Commerce Hoover and allowed the full $250,000 requested for export industries. This fund is to be used principally in an effort to assist the twelve chief export industries. In addition, the committee approved the full amounts which Secretary Hoover asked for the Bureau of Standards as follows: Testing structural materials, $50,000; industrial research, including elimination of waste, $100,000; standardization of equipment, $100,000.

These items are included in the second deficiency appropriation bill, which was reported to the House on May 18.

Work of the Power Commission

With the idea of conserving the time of the three secretaries who constitute the Federal Power Commission, it has been decided that in the future hearings will be held in the first instance by the executive secretary, who will render a preliminary decision. If those requesting a hearing are dissatisfied with this decision, appeal may be taken to the commission. In that event an agreed statement of fact, if such can be reached, is to be presented to the commission, together with a digest of the arguments for and against the matter at issue. Briefs may be presented and a further opportunity for oral argument will be given in each case by the commission. If no appeal is taken, the preliminary decision is to be presented by the executive secretary to the commission for approval.

Beginning with June the Federal Power Commission will meet regularly on the first and third Mondays of each month. At the request of the Secretary of War regular meetings have not been held since the change in administration.

The following order of business will be observed at future meetings: Findings with respect to effect of power projects upon navigable waters or reservations; authorizations for issuance of permits and licenses and restoration to entry of lands in power-site reserves; legal opinions; matters of policy or of general administration; appeals; formal orders; regulations; legislation.

Pending authorization by Congress that will permit the employment of properly qualified engineers and accountants, the commission will make no effort to undertake valuations of projects already constructed. This course of action was decided by the commission at its meeting on March 20. It is of the opinion that a satisfactory valuation of hydroelectric properties requires a personnel with specialized technical training and experience. Besides the chief engineer, whose other duties will not permit his undertaking the details of the work, persons with the necessary qualifications have not been assigned to the commission. The departments that are authorized to transfer personnel do no work of this character, and as a result they have no personnel qualified for these duties. In cases of this kind a license is to be issued under the condition that valuations shall be made at the earliest practicable date, and that they shall be determined as of the date of the license.

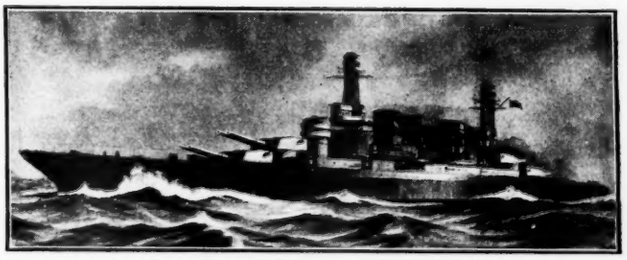
One Hundred and Eighty Thousand Horsepower in One Ship

The United States battle-cruiser "Ranger," shown here as it will appear when placed in service, is an interesting departure from usual American naval practice in several ways. The most notable feature of the vessel is the tremendous amount of power that will be concentrated in one hull. The electric system of propulsion is to be used, although that has become so common nowadays in naval construction as to be scarcely, in itself, a feature. The driving machinery, which is of Westinghouse make, will develop 180,000 hp., or more than that of any other ship in existence.

This great amount of power will enable the vessel to reach a speed of 33 knots, which is remarkable in view of the ship's length of 874 ft. and displacement of 43,500 tons. The "Ranger" is one of six battle-cruisers of this type now under construction.

Aside from the power and speed of these vessels their armament is of no small interest, as each ship is to carry eight 16-in. guns and will have good armor protection.

Practically all capital ships built for the United States Navy in the past have been designed for a speed of 21 knots; so it may readily be seen that the "Ranger" and her sister ships represent a radical step in naval construction.



AN 874-FT. BATTLE-CRUISER DESIGNED FOR A SPEED OF THIRTY-THREE KNOTS

New Hydro Plants Beginning Operation in California

Seattle is soon to begin operation of two more electric-power units costing about $1,000,000 each, besides opening a smaller hydro-electric unit preliminary to development of the Skagit power site.

The new unit of the Lake Union steam plant, generating electricity with oil as fuel, will be completed shortly, according to Light Department officials, making the plant one of the finest in the continent.

In June the third unit of the Cedar Falls hydro-electric plant will go into operation, including a new pipe line of wood stave and steel.

Late in June or early in July the Newhalem plant of the Skagit development will be generating power, but this small plant will be used only to supply power for the construction work on Gorge Creek, the first plant of the chain of power stations planned on the Skagit River. When the Gorge Creek station is completed and begins delivering power to Seattle, the current from the Newhalem plant also will be turned into the Seattle transmission line. Each of the two new major units will develop 20,000 hp. or about 15,000 kw.

The Lake Union steam plant is used as a reserve source of power during the peak load each night, and especially during the summer when the water supply at Cedar River diminishes. Including the new unit, it has cost $2,175,000 and will have a capacity of 35,000 kw., or about 46,000 hp.

The Gorge Creek power station, in the Skagit project, will not deliver power to Seattle until the late summer or fall of 1923.

On May 7, 60,000 hp. was added to California's power through the opening of the new plant of the Great Western Power Co. on the north fork of the Feather River. The new plant, when enlarged to its ultimate capacity, will be the second largest project of its kind in the West. Its present power output is estimated by electrical engineers to represent the constant labor of 480,000 men, according to the formula that one horsepower represents the labor production of eight men. San Francisco, Oakland, Sacramento, Napa, Martinez, Santa Rosa, Petaluma and twenty-two other cities will draw on this additional supply of power. It will be used for mining, industrial, domestic and commercial purposes; for lighting and for cooking and heating, gold dredging and for railroads.

Stoker Limitations with Mid-West Coals

In a paper presented at the Spring meeting of the A. S. M. E., Chicago, May, 1921, John E. Wilson stated that the average cost of repairs to a number of chain-grate and underfeed stoker installations has been found to be 3.7 and 5.4 cents, respectively, the cost of repairs to the fans and air ducts required by underfeed stokers being included in the latter figure. The chain-grate plants mentioned will burn, on an average, 30 lb. of coal per square foot of grate surface per hour, while the underfeed stoker plants will average 550 lb. per retort per hour.

The quantity of steam required for driving stokers, also for draft fans, which should in all cases be charged against stoker operation when considering these costs, will vary from 0.3 per cent on stokers using natural draft to 3 per cent where induced or forced draft is used.

The following figures are those of a representative heat balance from boilers equipped with natural-draft chain grates burning Illinois coal:

	Per Cent
Heat absorbed by water	70.0
Heat loss due to dry chimney gases	14.5
Heat loss due to combustible matter in the ash	5.5
Heat loss due to burning of hydrogen	5.0
Heat loss due to incomplete combustion of carbon	1.5
Heat loss due to moisture in coal	0.5
Heat loss due to moisture in air	0.5
Heat loss due to radiation and unaccounted for	2.5
Total	100.0

It will be seen from this that the greatest loss from any two combined sources is from the dry chimney gases and carbon in the ash.

It has been pointed out in a number of instances that small plants burning from 50 to 150 tons of coal a day operate very economically as compared with larger plants in the same district. This is due in a large measure to the fact that the supply of coal for the small plant is obtained from one source. Under these conditions the furnace design may be changed to suit the coal used, and the most satisfactory method of operating the stokers can be determined and adhered to. The coal for the large plant, on the other hand, is secured from a number of different sources and varies materially in quality and other characteristics. This requires very close attention in order to make the numerous changes necessary in the feed, air supply, etc., to suit the different coals.

One condition of primary importance to the stoker manufacturer which has come about in the past few years, is that the class of help in the boiler room has improved to such an extent that a piece of equipment is not barred from consideration on the ground that it requires intelligence to operate it.

The governors of the states in the Colorado River basin have presented a resolution to Congress pointing out that Arizona, California, Colorado, Nevada, New Mexico, Utah and Wyoming, by appropriate legislation, have authorized the appointment of commissioners representing these states for the purpose of entering into an agreement between these states and the United States respecting the future use and disposition of the waters of the Colorado River and of its tributaries. The governors urge that Congress provide for the appointment of commissioners on behalf of the United States.

A fund of £11,407 in honor of James Watt has been raised in England. It is proposed to give £5,000 to Birmingham University to establish a Chair of Research in Mechanical Science, and later to accumulate a fund large enough to provide a memorial building, which would serve as a meeting place and library for technical societies.

Enormous quantities of shale exist in Bulgaria, according to a recent Commerce Report. It is said that this shale is of high quality and forms the greatest deposit yet discovered.

New Publications

DISCUSSION OF THE NATIONAL ELECTRICAL SAFETY CODE. Handbook No. 4. Published by the Department of Commerce, Bureau of Standards, Washington, D. C., 1921. Cloth 5 x 7½ in. Price, 50 cents.

This handbook contains the revised discussion of the National Electrical Safety Code. It forms a companion volume to Handbook No. 3, which contains the revised edition of the rules themselves. The discussion of the rules has been considerably amplified and many items and suggestions included that are not involved in the rules themselves, but throw light on good practice and matters immediately related to the subject of the rules. In addition, it aims to explain and interpret the rules and to give in many cases the reasons that have prompted the inclusion of particular rules in the Electrical Safety Code. Illustrations have been provided to show the intent and application of a number of rules. The discussion of the rules follows the order of the rules themselves and is segregated into the four parts dealing, respectively, with stations and substations, overhead and underground lines, utilization apparatus, and operation of utility stations and lines. There is also a supplementary section dealing with the subject of grounding. The introduction to the discussion includes the results of a survey of pole lines made jointly with the committee on safety rules of the National Electric Light Association in the summer of 1919.

Handbook No. 4 may be secured from the Superintendent of Documents, Washington, D. C.

Personals

H. V. Coes has been appointed manager of the new branch office of Ford, Bacon & Davis that was recently opened in Philadelphia.

John W. Merton, formerly with the Hooven, Owens, Rentschler Co., Hamilton, Ohio, as engineer in charge of the Diesel department, is now with the New York Shipbuilding Corporation, Camden, N. J.

Edmund S. Glauch has opened an office as mechanical engineer in Harrisburg, Pa. He was previously assistant mechanical engineer for the Harrisburg Foundry and Machine Works.

R. K. Stockwell, until recently general sales manager of the Robins Conveying Belt Co., New York City, has been appointed British engineering manager for that company, and has taken up headquarters in London, England.

C. W. Koiner, for the last thirteen years general manager of the municipal light plant of Pasadena, Cal., and for the last year also city controller, has been appointed the first city manager of Pasadena, which has recently adopted the city manager form of government.

Louis T. Klauder, Pennsylvania Building, Philadelphia, Pa., has been appointed consulting engineer for the city of Reading, Pa., in connection with the design and construction of a new Maiden Creek Power Plant for the city. Design and construction are to be started immediately.

Society Affairs

New Jersey State Association, National Association of Stationary Engineers, will hold its annual convention June 4-5 at Central Hall, Newark Ave., Jersey City, N. J.

The Canadian Association of Stationary Engineers is to hold its annual convention and a "Power Exhibition" at Ottawa June 28-30. The convention secretary may be reached at 71½ Sparks St., Ottawa.

The Detroit Engineering Society recently elected the following officers for the ensuing year; president, M. W. Taber; first vice-president, J. R. McColl; second vice-president, A. L. Trout; secretary-treasurer, O. H. Dawson.

The American Order of Steam Engineers will hold its 35th annual national convention in Odd Fellows Hall, Allentown, Pa., June 6-10. Information may be obtained from Andrew Lauterbach, 4726 North Warnock St., Philadelphia, Pa. An extensive exhibit will be held.

New York State Association, National Association of Stationary Engineers, will hold its annual convention June 9-11 at the McAlpin Hotel, New York City. There will be a large exhibit in connection with the meeting. Further details may be had from T. Condon, 102 Seventh Ave., New York City.

The Central States Exhibitors' Association of the National Association of Stationary Engineers is holding a series of important conventions, with exhibits, as follows: Chicago, Ill., June 8-10; Cincinnati, Ohio, June 16-18; Anderson, Ind., June 23-25; Detroit, Mich., July 6-8; Wausau, Wis., July 20-22; St. Paul, Minn., Aug. 10-12. The secretary-treasurer is Charles H. Fiske, 431 South Dearborn St., Chicago.

American Institute of Electrical Engineers, at its annual business meeting in New York on May 20, announced the election of the following officers: President, William McClellan, Philadelphia, Pa.; Vice-presidents, Walter A. Hall, Boston, Mass.; N. W. Storer, Pittsburgh, Pa.; William A. Del Mar, Yonkers, N. Y.; C. G. Adsit, Atlanta, Ga; John C. Parker, Ann Arbor, Mich.; F. W. Springer, Minneapolis, Minn.; H. W. Eales, St. Louis, Mo.; Robert Sibley, San Francisco, Cal.; O. B. Coldwell, Portland, Ore.; F. R. Ewart, Toronto, Ont. Managers, A. G. Pierce, Pittsburgh, Pa.; R. B. Williamson, Milwaukee, Wis.; Harlan A. Pratt, New York, N. Y. Treasurer, George A. Hamilton, Elizabeth, N. J. (re-elected). F. L. Hutchinson remains the secretary.

Business Items

The France Packing Co. has moved from Grand Central Palace to 30 Church St., New York City.

Rickard & Sloan, 25 Spruce St., New York City, industrial and technical advertising specialists, have changed their corporate name to Rickard & Co., Inc.

The Buffalo Forge Co., Buffalo, N. Y., is remodeling its entire fan shop to such an extent that, when completed, it will cover 64,235 sq.ft. including 23,873 sq.ft. of assembling galleries.

The M. J. Dougherty Co., piping fabricators and contractors of Philadelphia, has changed the address of its Atlanta representative's office to Candler Annex. The office is in charge of Messrs McKee and Wright.

The Hauck Manufacturing Co., Brooklyn, N. Y., manufacturer of oil burners, oil forges, oil-burning appliances, etc., has elected as its president Henry T. Gerdes, mechanical engineer and manufacturer, of New York; first vice-president, M. C. Hauck; second vice-president, A. B. Hauck; third vice-president, H. H. Kress; treasurer, A. H. Stein; secretary, J. Lutz. Mr. Gerdes succeeds the late Arthur E. Hauck as president.

Trade Catalogs

The Lunkenheimer Co., Cincinnati, Ohio, has issued a small booklet on the subject of its line of "Ferrenewo" valves.

The Anderson Foundry and Machine Co., Anderson, Indiana, is distributing a pamphlet describing its new type K Surface-ignition oil engine.

The Carlyle Johnson Machine Co., Manchester, Conn., has prepared a Stock List of its slow-speed line of standard friction clutches (for speeds up to 250 r.p.m.) for immediate shipment.

The Donnelly Systems Co., 9 Murray St., New York City, is distributing a new bulletin on the one-pipe fractional vapor vacuum heating system and describing the company's fractional vapor regulator, packless radiator valve and weight-controlled vacuum air valves.

FUEL PRICES

BITUMINOUS COAL

The following table shows the trend of the spo steam market in various coals (mine-run basis, f.o.b mines):

Coal	Market Quoted	May 17 1921	May 24 1921
Pocahontas,	Columbus	$3.60	$3.50
Clearfield,	Boston	2.40	2.00@2.85
Somerset,	Boston	2.90	2.25@3.50
Pittsburgh,	Pittsburgh	2.10	2.00@2.20
Kanawha,	Columbus	2.30	2.50@2.75
Hocking,	Columbus	2.20	2.25@2.50
Pittsburgh No.8	Cleveland	2.30	2.15@2.30
Franklin, Ill.,	Chicago	3.25	2.75@3.50
Central, Ill.,	Chicago	2.60	2.25@3.00
Ind. 4th vein,	Chicago	3.10	2.75@3.50
Standard,	St. Louis	1.90	1.75
West Ky.,	Louisville	2.00	1.85@2.50
Big Seam,	Birmingham	2.95	2.85@3.10
S. E. Ky.,	Louisville	2.60	2.50@2.65

DIESEL FUEL OIL

New York—On May 26, Port Arthur light oil, 23@25 deg. Baumé, 5½c. per gal. 30@35 deg., 6½c. per gal. f.o.b. Bayonne, N. J.

Chicago—May 21, for 24@28 deg. Baumé, 55@60c. per bbl.; 32@34 deg., 2@2½c. per gal. In tank cars f.o.b. Oklahoma refinery, or freight adjusted

St. Louis—May 21, prices f.o.b. cars, tank lots; 24@26 deg. Baumé, 65c. per bbl.; 26@28 deg., 70c.; 28@30 deg., 80c.; 32@34 deg., 3c. per gal.

Baltimore—On April 23, f.o.b. refinery; in tank-car lots; 24+ deg. Baumé, 6¼c.; 26@28 deg., 6¾c.; 30@34 deg., 7¾c.; 14+ deg., $1.95 per bbl.; 16@20 deg., $1.85 per bbl.

Pittsburgh—On May 23, fo.b. refinery; Pennsylvania, 36@40 deg., 4¼c. per gal.; Oklahoma, 24@30 deg., 50c. gas oil, 32@34 deg., 2c. per bbl., 36@38 deg., 2¼ c. 38@40 deg. 2¼c.

Philadelphia—On April 12; 26@28 deg. Baumé, Oklahoma, 85c. per bbl.; 30@34 deg., Oklahoma (group 3), 3c. per gal.; 16@20 deg., Seaboard, 5c. per gal.

Cincinnati—April 23, for 22@28 deg. Baumé, 7.34 per gal.

Cleveland—April 23, for 22@28 deg. Baumé, 5½c per gal.

PROPOSED WORK

Vt., Searsburg—The New England Power Co., 35 Harvard St., Worcester, Mass., plans to build a hydro-electric plant along the Deerfield River here. Plant to have a generating capacity of 8,000 hp. About $1,500,000.

Mass., Beverly—The School Bd. plans to build a high school, etc. About $750,000. S. H. Chane, Supt. Architect not yet selected.

Mass., Clinton—The Connecticut River Power Co., 35 Harvard St., Worcester, will alter the power house here. About $50,000. Work will be done by day labor.

Mass., South Hadley—Mt. Holyoke College had preliminary plans prepared for a 2 story science building. About $300,000. Allen H. Cox, 114 State St., Boston, Archt.

Mass., Springfield—The Y. M. C. A. College, 287 Hickory St., plans to build an addition to college including science and dormitory buildings, etc. About $2,000,000. Architect not yet selected.

R. I., Pawtucket—The J. S. White Co., 22 Dexter St., is in the market for one air compressor, capacity 175 ft. per min., with motor and air compressor. (Used).

R. I., Providence — The Narragansett Electric Lighting Co., Turks Head Bldg., will soon award the contract for a 2 story, 114 x 172 ft. garage and boiler house on Melrose St. About $200,000. Jenks & Ballou, 735 Grosvenor Bldg., Archts. and Engrs.

Conn., Hartford—The First School Dist. will receive bids until June 27 for a 3 story, 110 x 140 ft. addition to the Brown School on Talcott St. A steam heating system will be installed in same. About $500,000. I. A. Allen, Jr., Inc., 904 Main St., Archt.

Conn., Stamford—The Bd. Educ plans to build a 2 story high school addition on Forest St. About $300,000.

N. Y., Buffalo—The Buffalo Box Factory, 30 Scoville Ave., is in the market for a 440 volt, 3 phase, a.c. 60 cycle electric hoisting machine to lift 1 ton about 15 ft., speed about 15 ft. to 25 ft. per min., chain or cable with hook. (Used and in good condition, preferred).

N. Y., Buffalo—The city will receive bids until June 3 for a transmission line to the Col. F. G. Ward pumping station—Massachusetts Ave. pumping station, 10,500 ft. of park cable, 2 lines. Ward Station, Lake View Ave. storehouse 1,500 ft. 1 line. G. C. Andrews, Water Comr. S. Firestone, 59 South Ave., Rochester, Consult. Engr.

N. Y., New York — (Manhattan Boro) The Knights of Columbus, Bway and 46th St., plans to build a 7 or 10 story building on 57th St. between Bway and 8th Ave. About $2,000,000. W. L. Larkin, Supreme Dir. Architect not yet selected.

N. Y., New York — (Manhattan Boro) The Realty Managers Inc., c/o Moore & Landseidel, Archts. and Engrs., 3rd Ave. and 148th St., is having plans prepared for a 5 story apartment house on Davidson Ave. and Fordham Rd. About $450,000.

N. Y., New York — (Manhattan Boro) D. Zipkin, 180 Bway., plans to build a 16 story, 100 x 150 ft. garment centre loft building at 247-263 West 37th St. About $1,000,000. Architect not yet selected.

N. Y., New York — (Manhattan Boro) J. Zubow, c/o Bloodgood & Sugarman, Archts. and Engrs., 17 East 49th St., had plans prepared for a 12 story office building at 343 Madison Ave. About $500,000.

N. Y., Syracuse—The city plans to build 2 junior high schools, one to be located on Midland Ave. About $500,000 each. H. C. Allen, City Engr.

N. J., Atlantic City—The Atlantic City Hospital plans to build a 2 story addition to its hospital on Ohio Ave. About $500,000. Architect not yet selected.

N. J., Trenton — The Mechanics Natl. Bank, State and Warren Sts., plans to build an 11 story, 30 x 80 ft. office and bank building on West State St. About $1,000,000. Architect not yet selected.

N. J., Trenton—The Trent Theater Building Co., North Warren St., plans to alter its theater. About $500,000. W. H. McElfatrick, 701 7th Ave., New York City, Archt.

Pa., Altoona—The Bd. Educ., will soon award the contract for a 1 story, 100 x 120 ft. power and heating plant on 6th Ave. About $200,000. W. N. Decker, 6th Ave. and 15th St., Secy. Wood, Hulse, Yates & Co., Inc., Woolworth Bldg., Archts. and Engrs.

N. C., Greensboro—The Jefferson Standard Life Insurance Co. plans to build a 15 story, 115 x 187 ft. office building including a vapor heating system on West Market St. About $1,000,000. C.C. Hartman, Archt.

Ga., Calhoun—L. N. Legg, manufacturer of second hand machine tools is in the market for 18 ft. x 72 in. high pressure tubular boilers, two 9 ft. Chambers dry pans, latest model, 16 x 36 Hamilton Corliss engine and a gasoline locomotive. (second-hand).

Fla., Moore Haven—The city will soon award the contract for an addition to the light plant including a 150 hp. oil engine direct or belt connected to a 125 k.v.a. generator. About $23,000. R. O. Baker, Clk. T. E. Frederick, Engr.

Tenn., Memphis—The city voted $1,500,000 bonds to construct an auditorium on Main and Poplar Sts. C. O. Pfeil & G. Awsumb, U. & P. Bank Bldg., Archts.

O., Cleveland—The city will receive bids until June 3 for furnishing 250 (750 watt) multiple transformers, etc. P. S. Beckwith, City Hall, Engr.

O., Cleveland—J. Laronge, 214 Williamson Bldg., plans to build a 12 story, 40 x 130 ft. commercial building including a steam heating system on East 12th St. North of Euclid Ave. About $300,000. W. S. Longee, 500 Marshall Bldg., Archt.

O., Columbus—The Bd. Educ. is having plans prepared for a 2 story high school to be known as the South High School. About $1,000,000. Richards, McCarthy & Bulford, 384 East Broad St., Archts.

O., Piqua—The B. F. Keith Co., 19 Walnut St., Cincinnati, plans to build a 3 story theater on Main St., here. About $800,000. L. R. Lipson, Mgr. Architect not yet selected.

Ind., South Bend—The A. A. O. of Scottish Rite is having plans prepared for a 6 story, 132 x 165 ft. Masonic temple including a steam heating system. About $500,000. R. G. Schmidt & Co., 154 West Randolph St., Chicago, Ill., Archt.

Mich., Detroit—The Bd. of Fire Comrs., Wayne and Larned Sts., will receive bids until June 6 for furnishing six multi stage horizontal centrifugal pumps with flexible coupling connected to slip ring a.c. motors, to have a capacity of 2,500 gal. per min., each against a head of 700 ft. including suction lift of 15 ft. 10 in. suction and discharge nozzles and 1200 Synchronous motors, etc. Smith, Hinchman & Grylls, 710 Washington Arcade, Engrs.

Mich., Detroit — G. D. Mason & Co., Archts., 508 Griswold St., will receive bids until June 11 for a 4 and 6 story, 190 x 350 ft. masonic temple on Second Blvd. and Temple Ave., for the Masonic Temple Association. About $3,500,000.

Mich., Manistique—J. Christensen, City Clk., will receive bids until June 15 for a pumping station, tower, etc. Centrifugal pumps will be installed in same. Alvord & Burdick, 8 South Dearborn St., Chicago, Ill., Engrs.

Mich., Saginaw—The Union High School Dist. voted $500,000 bonds to construct a 2 story high school including a steam heating system. N. S. Spencer & Sons, 37 West Van Buren St., Chicago, Ill., Archts.

Ill., Chicago—The Bd. Educ., 650 South Clark St., is having plans prepared for a 3 story, 80 x 200 ft. addition to the Taylor Grade School including a steam heating system on 99th St. and Avenue J. About $600,000. John Christensen, Archt. John Howatt, Engr.

Ill., Chicago—The Bd. Educ., 650 South Clark St., is having plans prepared for a 3 story, 300 x 650 ft. high school including a steam heating system on Wilson Ave. and Kimball St. About $4,000,000. John Christensen, Archt. John Howatt, Engr.

Ill., Chicago—A. M. Rutenberg, Archt., 105 North Clark St., had preliminary plans prepared for a 3 story, 160 x 175 ft. apartment house including a steam heating system. About $500,000. Owner's name withheld.

Ill., Chicago—S. J. Schaeffer, 155 North Clark St., is having plans prepared for a 6 story, 85 x 150 ft. apartment house including a steam heating system on Kenmore and Balmoral Aves. About $600,000. E. E. Hall, 123 West Madison St., Archt.

Ill., Wheaton—Wheaton College plans to build several college buildings. About $500,000. C. H. Blanchard, Pres. King & Campbell, 36 West 40th St., New York City, Archts. and Engrs. Noted May 23.

Ia., Davenport—The Nichols Wire & Sheet Co. is in the market for an air compressor for dry pipe sprinkler system—capacity 8 to 11 cu.ft. per stroke. (New or Used).

Mo., Independence — The city plans to build an addition to electric light plant here. About $35,000. Black & Veatch, Mutual Bldg., Kansas City, Engrs.

Mo., St. Louis—The Concordia Seminary, c/o J. H. Fritz, Pres., Jefferson St. and Winnebago Ave., plans to build several college buildings including a steam heating system at Hi Pointe. About $2,500,000.

Mo., St. Louis—The J. L. Sprague Real Estate Co., Chemical Bldg., had plans prepared for a 5 story, 145 x 309 ft. hotel including a steam heating system. About $500,000. Edward Lantz, Post Dispatch Bldg., Archt.

Col., Aurora—The Constructing Quartermaster, c/o Fitsimmons Genl. Hospital, will receive bids until June 15, for a number of buildings including a pump house, etc.

Ore., Portland—D. B. McBride and R. F. Wassell, Royal Bldg., are having plans prepared for a 9 story, 92 x 100 ft. apartment house on 6th and Madison Sts. About $450,000. Carl L. Linde, 637 Piltock Bldg., Archt. and Engr.

Ore., Salem—The State Bd. of Control will receive bids until June 2 for a steel tank, deep well pump, steel tower, etc.

Cal., Los Angeles—The Bd. Educ. will soon receive bids for a high school on 4th and Fickett Sts. About $324,000. Hunt & Burns, 701 Laughlin Bldg., Archts.

Que., Montreal — The Kingsbury Footwear Co., 679 La Salle Ave., will soon award the contract for a chimney and installation of two 100 hp. steam boilers. About $130,000. I. Lafahvre, Engr.

Que., Quebec City—The Banque Nationale, St. Peter St., will soon award the contract for a 5 story bank and office building. About $300,000. A. Gaudry, c/o Owner, Archt.

Ont., Petrolea — The Canadian Oil Co. plans to construct a boiler house including the installation of 4 new boilers, etc. About $50,000.

Ont., Toronto—The Bd. Educ., 155 College St., has purchased a site on Wellesley St. and plans to build a 2 story collegiate building including a steam vacuum heating system, ventilation equipment, etc., on same. About $300,000. W. W. Pearse, Archt.

CONTRACTS AWARDED

R. I., Newport — The School Building Com. has awarded the contract for installing a steam heating system in the proposed 3 story, 100 x 146 ft. high school on Central St., to M. A. Dame & Son Co., Lynn, Mass., at $46,350. Noted Feb. 15.

Conn., Waterbury—The Citizens & Manufacturers Natl. Bank has awarded the contract for a 1 story, 85 x 100 ft. bank including a steam heating system on Leavenworth St. and Kendrick Ave., to the Leavenworth Co., Leavenworth St. About $300,000.

N. Y., Buffalo—The State Hospital Comn., Capitol Albany, has awarded the contract for remodelling the heating plant at the Buffalo State Hospital here, to John W. Danforth, 70 Elicott Sq., at $54,399. Noted April 26.

Pa., Scranton—The Scranton Dress Co., c/o Lockwood Green & Co., Archts. and Engrs., 101 Park Ave., New York City, has awarded the contract for a 1 story factory including a steam heating system at 100 South Webster Ave., to F. W. Ferguson, Paterson, N. J. About $350,000.

Ga., Atlanta—J. A. Manger has awarded the contract for a 9 story addition to the Cecil hotel on Cone and Duckie Sts. to the Griffin Constr. Co., Candler Bldg., at $300,000.

O., Warren—The Bd. Educ. has awarded the contract for the installation of a heating and ventilating system in the 2 proposed Junior high schools, to Barnes & Clark, Newcastle, Pa., at $36,846 each. Noted May 10.

O., Youngstown—Henry Dietz, Imperial Hotel, has awarded the contract for installing a plumbing and heating system in the proposed 5 story, 80 x 120 ft. addition to hotel on Boardman St., to the C. F. Henry Co., 645 Market St.

Ill., Sesser—The Old Bond Coal Co., 332 South Michigan Ave., Chicago, will build a 1 story, 50 x 83 ft. boiler house, here. About $50,000. Work will be done by day labor.

Wis., Milwaukee—The Sewerage Comn. has awarded the contract for installing boilers in the power plant on Jones Island, to the Heine Safety Boiler Co., 38 South Dearborn St., Chicago, Ill., at $113,300; for Foster superheaters, to the Power Specialty Co., 111 West Monroe St., Chicago, Ill., at $15,000; for economizers and induced draft equipment, to B. F. Sturtevant Co., 530 South Clinton St., Chicago, at $65,114; for stokers and forced draft-equipment, to the Westinghouse Electric & Mfg. Co., 111 West Washington St., Chicago, Ill., at $39,920. Noted May 3.

Kan., Plainsville — The Glosbasch Grain Co. has awarded the contract for a grain elevator building to be electrically equipped to the Farmers Silo Co., 1822 Main St., Kansas City, Mo.

Wyo., Basin—The town, c/o F. L. Cox, Clk., has awarded the contract for one 120 k.v.a., 90 kw., 900 r.p.m., 3 phase, 60 cycle, 1150 volt general electric generator and one 3 kw., 125 volt, 1700 r.p.m. general electric exciter, to Hendrie & Bolthoff Mfg. & Supply Co., 1639 17th St., Denver, Col., at $1,674; and for 1 Fairbanks-Morse, 4 in., 3 stage heavy duty pump—5 in. suction and 4 in. discharge, direct connected with flexible coupling to Fairbanks-Morse, 75 hp. 1800 r.p.m. 60 cycle, 3 phase, 220 volt, a.c. motor, to the Casper Supply Co., Casper, Wyo., at $2,175.

Ont., East London (London P. O.)—The Pub. Utilities Comn., Hydro Bldg., London, has awarded the contract for a 2 story, 50 x 85 ft. hydro-electric sub-station here, to the Hyatt Bros., 288 Egerton St. About $50,000. Noted Jan. 11.

Ont., Peterborough—T. H. Kennedy has awarded the contract for a 1 story, 50 x 100 ft. ice manufacturing plant, to C. J. Rutherford, 328 London St. About $35,000.

Ont., Sarnia—The School Bd. has awarded the contract for a 3 story high school including a steam heating and mechanical ventilation system on Wellington St., here, to P. H. Secord & Sons, 133 Nelson St., Brantford at $575,000.

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